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PRIL, 1939

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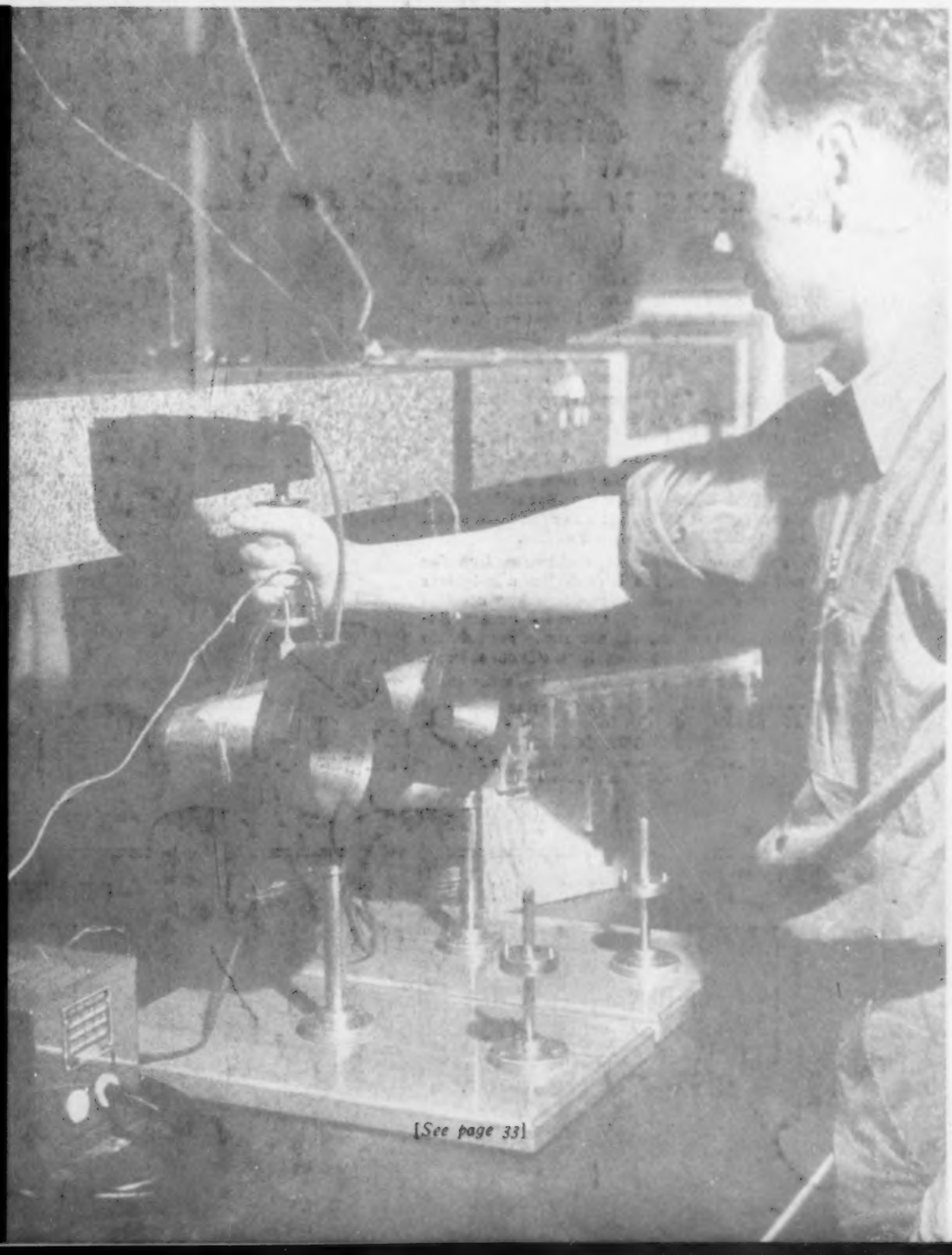

[See page 33]

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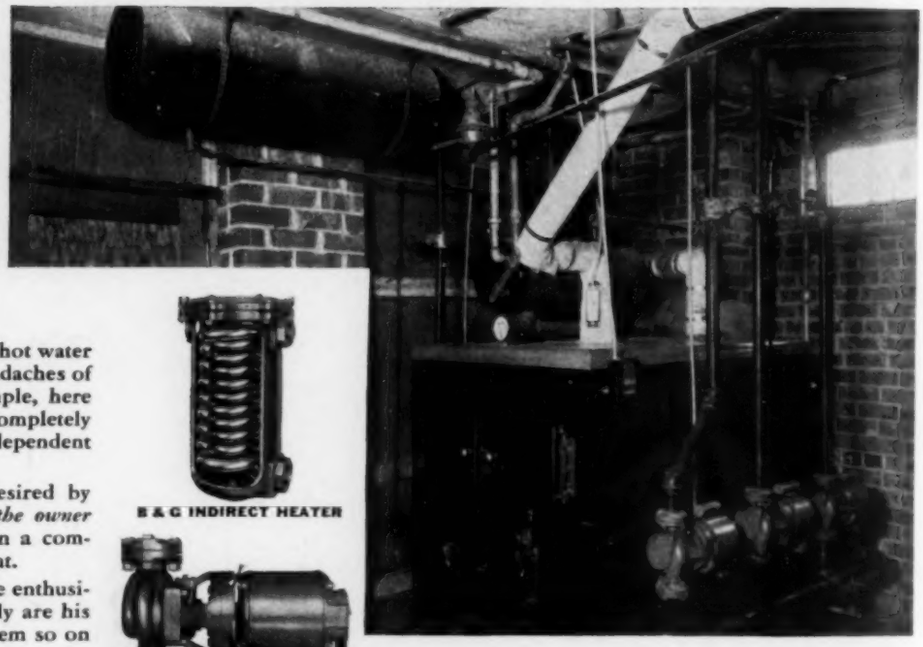

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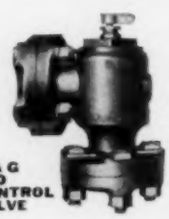

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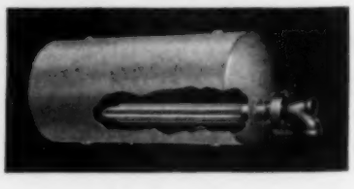


The Editor's Page

■ **Because of the** importance of propeller fans in the field of ventilation, the National Association of Fan Manufacturers has sponsored the formation of the Propeller Fan Manufacturers Association. A preliminary meeting was held at Pittsburgh, January 24, and a formal organization meeting in Detroit on February 28. At Detroit the association was launched with 20 charter members, a permanent chairman and secretary were elected, and a committee appointed to draw up by-laws and rules. At that time the next meeting was set for Detroit on April 4. . . . V. C. Shetler is secretary of the association and the headquarters office is at 2-255 General Motors Bldg., Detroit, Mich.

■ **Temperature** and Its Measurement—A symposium on temperature and its measurement in science and industry is to be held under the auspices of the American Institute of Physics probably next fall, although the exact dates are to be announced later. Primary purposes, according to present plans, are to co-ordinate a treatment of the subject in the sciences and in engineering, review principles, accumulate contributions for a comprehensive text, reveal the subject as an important branch of physics, and supply schools with information required for the improvement of curricula. . . . C. O. Fairchild, director of research of C. J. Tagliabue Mfg. Co., is chairman of the steering committee which is interested in hearing from those who want to take part in the symposium. . . . Address of the institute is 175 Fifth Ave., New York, N. Y.

■ **An interesting** and worthwhile project has been initiated in a number of cities—the Men Over Forty Club. It is a co-operative undertaking of men who have earned $4,000 or more a year, each without a present business connection and each over 40 years of age, who have pooled their labors, experience and intelligence in a self-organized, self-sponsored, self-managed and self-sustaining organization, the prime purpose of which is to obtain connections for its members. . . . The Chicago group is limited to 50 members. When active members secure positions, they become alumni and their places on the rolls are filled with new members.

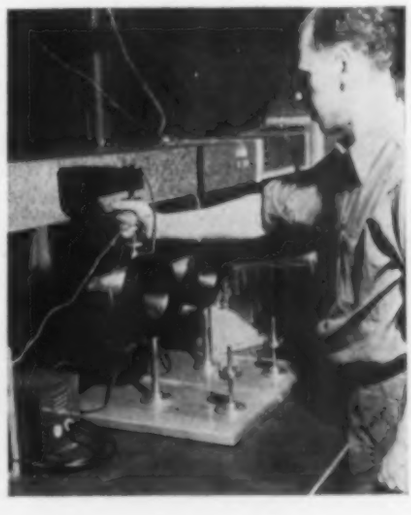
■ **Air Conditioning Symposium** — The papers presented at the air conditioning symposium held in Chicago November 22 are published in the February, 1939, *Journal* of the Western Society of Engineers, 205 W. Wacker Dr., Chicago, Ill., which has just been issued. Price of the *Journal* is 75c a copy. It includes The Development of Air Conditioning, by Willis H. Carrier; Engineering Aspects of Air Conditioning, by Charles S. Leopold; What Air Conditioning Means to the Architect, by Albert Buenger; Refrigeration as Applied to Air Conditioning, by John R. Hertzler; and Available Air Conditioning Equipment, by William B. Henderson. . . . The symposium was reported in the December, 1938, HEATING, PIPING AND AIR CONDITIONING, where the papers were summarized but not given in complete detail.

■ **The American Society** for Testing Materials, 260 S. Broad St., Philadelphia, Pa., has issued an index to its 870 standards and tentative standards as of January 1. The index is of service to any one wishing to ascertain whether the ASTM has issued standard specifications, test methods, or definitions covering a particular engineering material, and in locating the standards and the volumes where they appear. . . . Copies of the index may be had without charge from ASTM headquarters.

■ **Practically the entire** staff of the department of mechanical engineering, many of the research men, and some of the representatives of the building department, were on the job to put over the air conditioning conference at the University of Illinois last month, which was attended by 277 people from eleven states and the District of Columbia and devoted to the "fascinating and elusive subject of air conditioning," to use the words of Dr. A. C. Willard, president of the university. The sessions were well attended and the discussions lively. At the informal dinner meeting, Dr. Charles Sheard of the Mayo Clinic and the Mayo Foundation gave an address on the physiological responses of the body to its environment. Dr. Willard presided at the dinner meeting and among those at the speakers' table he introduced were Dr. and Mrs. Sheard, Dean M. L. Enger of the college of engineering, Prof. O. A. Leutwiler, head of the department of mechanical engineering, Mrs. Leutwiler, Dr. R. W. Keeton, of the college of medicine, Samuel R. Lewis, and E. B. Langenberg. A telegram of congratulations from J. F. McIntire, president of the ASHVE and a graduate of the U of I college of engineering, was read. . . . In addition to the daily sessions and the dinner meeting, conference attenders were given the opportunity to see the mechanical engineering laboratory with its many research and instruction set-ups. Among the exhibits that attracted attention were the new room heating test plant for studying steam, electric and hot water heating units under actual service conditions, and for air distribution studies employing warm air; a complete all year air conditioning apparatus; the warm wall test booth; and much other equipment and instruments. Inspection trips were also made to the air conditioned Lincoln Hall theater with its three zones of air distribution, evaporative condensers and other features, and to the warm air research residence. . . . The papers presented at the conference will comprise a circular which will be available next month. For a limited time, copies will be free to those who make direct application to the Engineering Experiment Station, University of Illinois, Urbana, Ill.

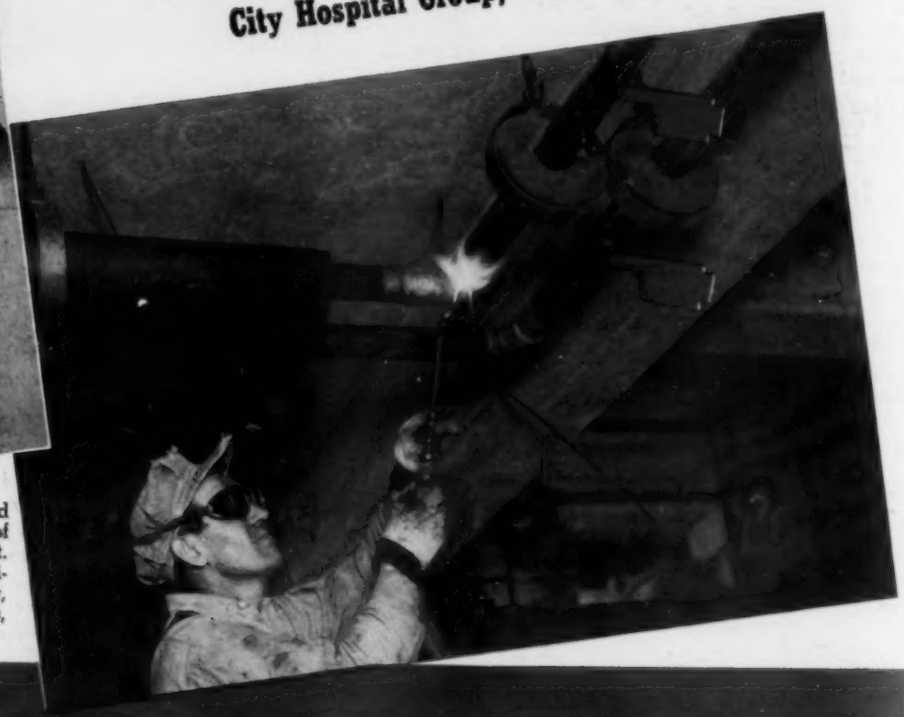
THE FRONT COVER

An observer at the University of Illinois sights a radiation pyrometer on the side of a sheet metal duct which is carrying warm air in order to determine the temperature of the duct surface in studying heat loss from ducts. The cones are the black body reference cones which are necessary for calibration. This apparatus is used in determining surface temperatures of walls, floors and ceilings as well as ducts, and in measuring the skin temperatures of human subjects.



FOR EIGHT CORROSIVE SERVICES

in the Malcolm A. Bliss Psychopathic Institute, City Hospital Group, City of St. Louis



Designed by the Bridge and Building Section of the Board of Public Service of the City of St. Louis. William C. E. Becker, Division Engineer, Albert Osburg, Chief Architect, John C. Ross, Mechanical Engineer.

BYERS WROUGHT IRON

The engineering approach to piping selection is well illustrated in the case of the Malcolm A. Bliss Psychopathic Institute, City Hospital Group, City of St. Louis. Study revealed corrosion as a definite threat in a number of lines where any failure would involve serious consequences and considerable replacement difficulties. These were medium and low pressure steam and steam return risers between the attic floor and basement ceiling; compressed air piping throughout; brine flow and brine return piping throughout; waste, soil, and vent lines above basement ceiling; cold water lines; ice water lines; and hot water trunk lines in attic and basement. Service records showed wrought iron to be the material with greatest life expectancy under these conditions. Byers Wrought Iron was used.

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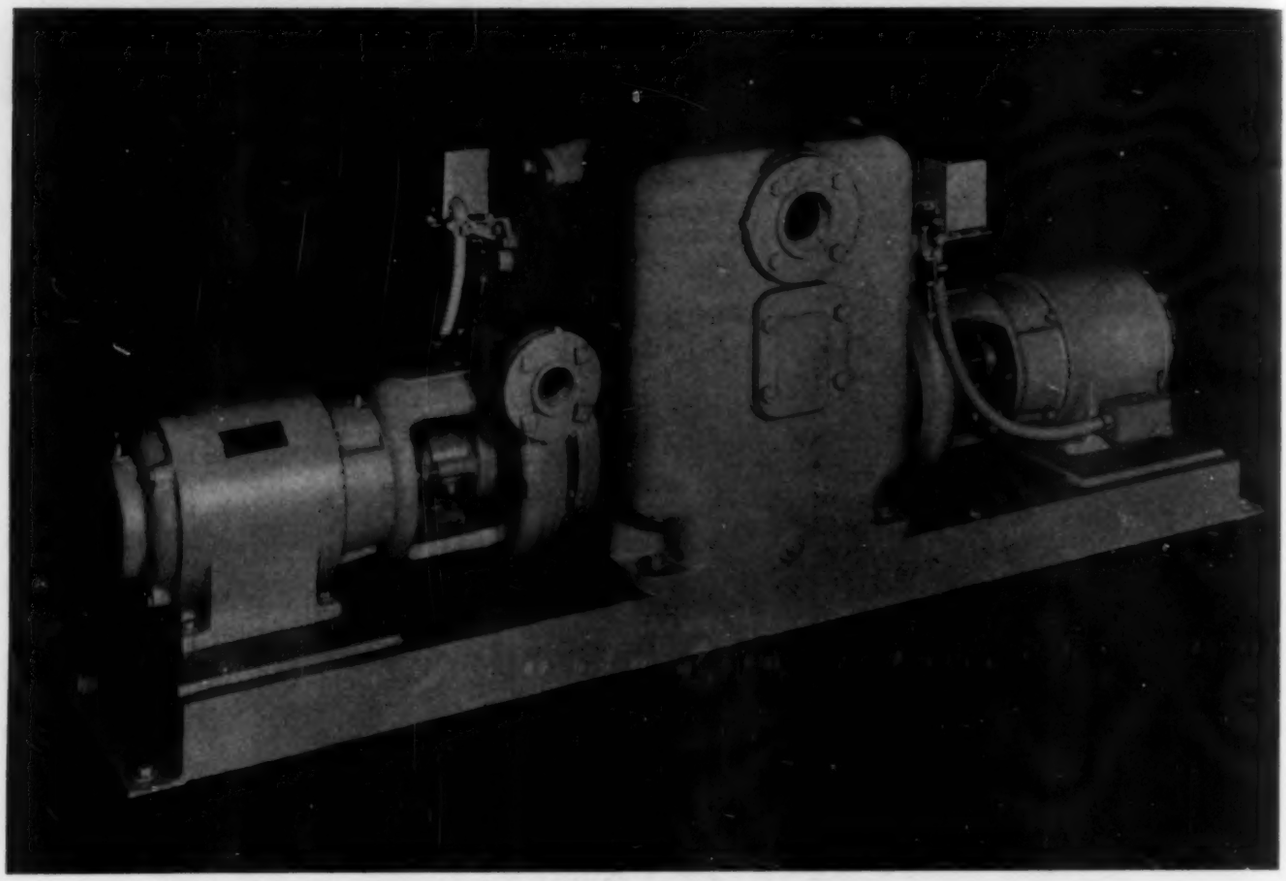
MAXIMUM EFFICIENCY FOR STEAM HEATING SYSTEMS ON HEATING, DRYING AND PROCESS WORK.



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Heating · Piping
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Vol. 11, No. 4 ● *April,* 1939

Heating the World's Largest Apartment Project

By Henry G. Schaefer*

THE Metropolitan Life Insurance Co. has acquired the site of the New York Catholic Protectory, embracing some 129 acres in the heart of Bronx County in New York City, and is proceeding with the construction of a development that will ultimately house more than 12,000 families, and will provide commercial centers catering to their wants, as well as recreational and amusement facilities. Believed to be the "world's largest" planned apartment community initiated and financed by private capital, the first section is scheduled for occupancy this fall.

The apartments (the bulk of which will be divided equally between three and four room suites) will house approximately 40,000 persons in 171 dwelling units averaging eight stories in height, and grouped in 51 entirely separate structures. Two theaters will provide entertainment, and outdoor recreational facilities are incorporated in the landscaping program, itself an aesthetic exploitation of the topographical features of the site. More than 9000 lineal feet of shop street-frontage will be available for commercial space for the convenience of the community. This space will be divided between the major shopping center, situated in the southeastern section of the development, and three regional centers which are designed to provide "corner grocery" convenience.

*Job Supervisor, Baker, Smith & Co., Inc.

Sketch of some of the buildings of the project

Insulated Walls for Economy and Comfort

The question of heating this mammoth project was given particular attention. The item of fuel cost is the largest single item in the maintenance cost of New York dwellings; hence, both initial investment in heating equipment and operating cost were carefully considered. The initial attack on the problem was directed against the major factors in heat loss, and the heat transmitting characteristics of the structure were studied. As a result, the conventional wall of plaster on waterproofed brick was revised to include an inch of insulation, with a vapor stop, between brick and plaster. Glass wool was selected as the insulator.

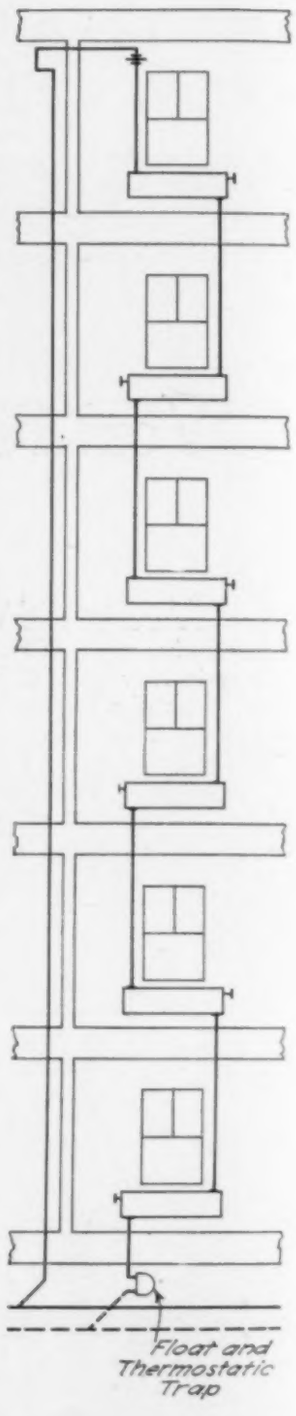
Probably the most important effect of the insulation of the structural walls will be the comfort provided. During the heating season, the discomfort that goes with low interior wall surface temperatures will be eliminated in that the differential in temperature between the exterior and interior surfaces of the outer walls will, of course, be much greater than in the case of uninsulated wall construction. Hence, due to the higher temperatures of the plaster, the absorption of the radiant body heat will be reduced, and with it the necessity of carrying higher air temperatures to offset this source of discomfort—a phenomenon capitalized in "radiant heating." This aspect of the installation promises even further fuel savings than those considered in the purely mathematical analysis of the matter. A further effect will be the protection of the apartments and their occupants from the direct summer sun heat. This advantage is an uncapitalized byproduct of the heating economies.

Having thus brought the heating load down to the practical minimum, the heating medium and the type of apparatus to dispense the heat came in for consideration. Further economies were sought, both in investment cost and in operation and maintenance. Space requirements, appearance, and life of the equipment were among the factors involved.

Sub-Atmospheric Steam for Heating

It was decided to use steam as the heating medium, under sub-atmospheric pressures during the bulk of the heating season and at atmospheric or slightly positive pressures during periods when extreme outside temperatures prevail. Group thermostatic control is to be used. As a permanently tight system is necessary to permit operation at the higher vacuums, this was made a fundamental requirement. The operating range is to be from a vacuum of 25 in. of mercury (133 F temperature) to 2 lb gage pressure (218 F).

The next step was selection of the type of radiator and piping installation, giving consideration to the tightness of the system, concealment of the radiation, and so forth. The solution decided upon was the use of a newly developed one pipe, looped, valveless, enclosed convector heating system which puts the kitchen and bathroom heat risers to work as conduits. The diagram illustrates the installation. The heat riser referred to is carried to the top floor ceiling, where it transfers to the adjacent radiator location (or to two, in the case of four and five room suites), thence descending through a series of convectors. Each cast iron convector constitutes an offset in the riser which alternates in its position from one side of the window to the other on successive floors, until it reaches the basement, where it is trapped into the return main. The dog leg offset provides a cushion for taking up expansion strains, the radiator and piping weight being supported by a bracket integral with the enclosure at the outlet end, thus distributing the weight of the installation floor by floor.



Loop method of piping

The convector enclosure is 4 in. wide, and the unit extends just far enough beyond the line of the window jamb to house the entering and leaving riser. A "one pipe" installation is thus had, with a single straight pipe connection between radiators. The specially designed radiator is tapped for a top inlet and a bottom outlet, eliminating elbows and elbow friction. The piping system is the subject of a pending patent application.

Dampers fitted into the convector enclosures to regulate or to stop the air flow give the room occupants control of the heat output within the maximum limit imposed by the control system, which latter provides the

The Metropolitan Life Insurance Co. is building in New York what is believed to be the "world's largest" planned apartment community initiated and financed by private capital. It will provide accommodations for over 12,000 families and is to include a major shopping center, local shopping centers, theaters, and complete recreational and amusement facilities . . . . The buildings, which average eight stories in height, will be heated by sub-atmospheric steam supplied from a central boiler plant. Building insulation will promote heating economy and comfort. The arrangement of the steam piping and the method of installing the convectors, described here, are of particular interest

"governor" that prevents overheating—that arch thief of fuel and comfort. Basement utility spaces are to be heated by the simple means of omitting covering from sufficient main and branch surface to provide the necessary heat emission, thus saving pipe covering and eliminating radiators at the same time.

Boiler Plant Designed for High Pressure

Once the foregoing fundamentals of the heating equipment had been determined, the matter of steam generation was considered. This also involved the consideration of the electric current requirements. Again, the insulation of the structures had a startling effect. In normal local apartment work, the ratio of heat to hot water load is approximately 120 to 42 sq ft EDR. Design and insulation contributed to reduce this ratio in the case of this development to 66 to 42 sq ft EDR. An extensive economic analysis of the situation directed the installation of a central high pressure boiler plant, built for 500 lb operating pressure so that it may be used for current generation if this should become desirable or necessary, but arranged for initial service for heating only at 100 lb pressure.

The boiler installation comprises four water-walled generators, each capable of supplying 100,000 lb per hr of steam, with overload capacity of 35,000 lb additional. The plant is being built for Bunker C fuel oil firing, but arranged for the provision of bunkerage, pulverizing and combustion of coal. A 275 ft radial brick stack will carry the products of combustion high enough so that no local contamination of the air will result.

Turbine driven forced and induced draft fans and boilerfeed pumps will contribute to the operating economy of the plant. The steam will be distributed at boiler pressure to reducing valve stations, each serving approximately 450 apartments. Successive stages of pressure reduction will prepare the steam for service in hot water generators, unit heaters for the stores, and the direct heating systems, all under thermostatic control.

The steam distributing system will be contructed of welded steel pipe with stainless steel bellows type expansion joints. The lines will be laid in concrete tunnels, with frequent manholes for access to valves and traps, and for inspection. The condensate will be returned to the boiler plant in a parallel system of piping laid in the same tunnel system.

Resistance of Materials to Vapor Transmission

L. V. Teesdale, senior engineer, U. S. Department of Agriculture Forest Service, Forest Products Laboratory, Madison, Wis., presented the accompanying table in his paper on condensation problems in modern buildings presented at the University of Illinois air conditioning conference last month.

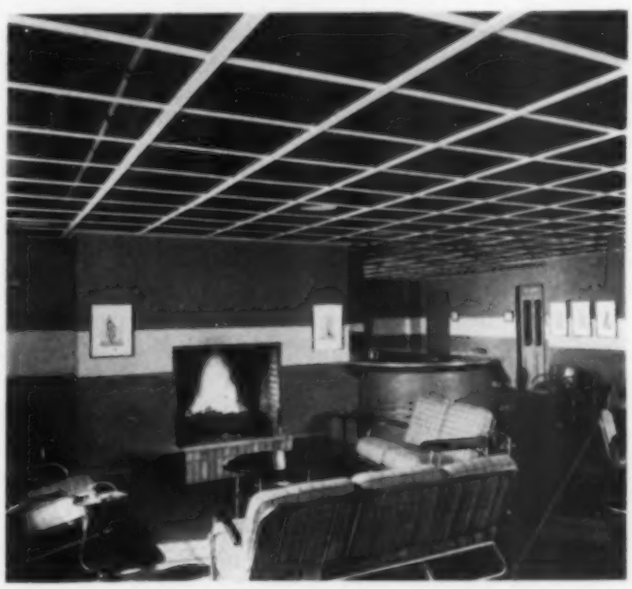
Comparative Resistance of Various Materials to Vapor Transmission

Material	Loss in Grains Per Sq Ft Per Hr	
Foil surfaced reflective insulation (double faced)	0.061	0.093
Roll roofing—smooth surface:		
40 to 65 lb per roll 108 sq ft	0.093	0.123
Asphalt impregnated and surface coated sheathing paper glossy surfaced:		
50 lb 500 sq ft roll	0.153	0.555
35 lb 500 sq ft roll	0.123	1.480
Duplex or laminated papers 30-30-30	0.990	1.850
Duplex or laminated papers 30-60-30	0.370	0.617
Duplex papers reinforced	0.493	1.480
Duplex paper coated with metal oxides	0.370	0.930
Insulation backup paper, treated	0.617	2.462
Gypsum lath with aluminum foil backing	0.061	0.277
Plaster—Wood lath	....	7.90
Plaster—Three coats lead and oil	2.650	2.770
Plaster—Three coats flat wall paint	....	3.080
Plaster—Two coats aluminum paint	....	0.831
Plaster—Fiberboard or gypsum lath	14.20	14.80
Slaters felt	3.700	18.50
Plywood—¼ in. Douglas fir, soy bean glue plain	3.080	4.620
Two coats asphalt paint	....	0.308
Two coats aluminum paint	....	0.930
½ in. 5 ply Douglas fir	1.920	1.975
¼ in. 3 ply Douglas fir, art. resin glue	3.080	4.620
½ in. 5 ply Douglas fir, art. resin glue	1.975	2.420
Insulating lath and sheathing—board type	18.50	24.65
Insulating sheathing, surface coated—3/16 compressed fiber board	....	3.640
1 in. insulating cork blocks	....	4.440
½ in. and 1 in. blanket insulation between coated papers	1.380	1.440
4 in. mineral wood—unprotected	....	20.950

Tests were made to determine the comparative vapor resistance of various papers and wall materials used in building construction. Samples were sealed in copper pans containing water and exposed in a room controlled at 80 F and 30 per cent relative humidity and weighed regularly for 90 days or more. The values obtained after the rate of loss became constant were calculated on a basis of grains of moisture lost per sq ft per hr.

This is only a partial list of the materials tested up to the present time, and as the tests are incomplete, it will be subject to change as required with further work. Many of the materials have been tested under actual exposure conditions in laboratory test house wall panels.

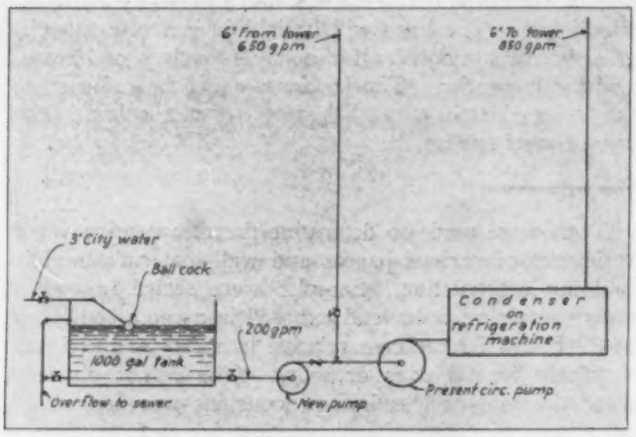
When a country club recently installed a lounge in its basement, it was desired to conceal as much as possible the radiators and piping which were on the ceiling. The trick was accomplished by installing a lattice ceiling to create an illusion of solidity



City Water Assists Cooling Tower When Wet Bulb Temperature is High

By John Howatt*

IN the spring of 1936 a 100 ton steam jet refrigeration unit was installed in a large office building for providing chilled water for air conditioning the space occupied by two of the tenants. As the building generates electric power for its own use and for other buildings in the block, and the exhaust steam is used for heating in winter, the steam jet refrigeration



On days of high wet bulb temperatures, cooler city water assists the cooling tower in furnishing water of the proper low temperature to the refrigeration condenser

equipment was designed to operate on the available 4 lb pressure steam and thus utilize exhaust steam in the summertime as well as in the heating season.

The building power plant has both non-condensing reciprocating engine and steam turbine driven electric generators, and there was an existing cooling tower for cooling the condensing water for the turbine. When the steam jet refrigeration installation for air conditioning was designed it was contemplated that during hot summer weather power would be generated with the non-condensing engines only, as it was recognized the cooling tower would not have enough capacity to furnish condensing water for both the electric generating and the air conditioning requirements in hot weather.

During the first season's experience with the air conditioning system, difficulty was encountered in getting condensing water from the cooling tower at a temperature low enough for the steam jet refrigeration unit. Part of the trouble was due to the fact that to keep down the cost of generating electricity, the turbine driven generator was operated even on hot days at the same time the air conditioning equipment was in use, thus overloading the tower. An investigation showed also that effectiveness of the cooling tower was impaired because of insufficient air motion through it, lack of water spray plates over part of the top, some of the spray plates were missing, and a large steam exhaust head was situated adjacent to one of the tower fan intakes. It was recommended that the practice of operating the turbine driven generator in severe weather when air conditioning was needed be stopped, and that the capacity of the cooling tower be increased by increasing the size of the fan units and replacing the missing spray plates and splash discs.

There was still another difficulty—that of high outside wet bulb temperatures. The cooling tower was designed on the basis of a maximum wet bulb of 75 deg, and in the particular locality the wet bulb temperature exceeds this value on an average of 10 to 20 days a year. It was to be expected, then, that even with a cooling tower of proper capacity carrying only the air conditioning load, some discomfort would be experienced by the occupants of the conditioned spaces on days when the outdoor wet bulb temperature was higher than the 75 deg for which the cooling tower was designed.

The accompanying sketch shows how this difficulty was economically solved by assisting the cooling tower; on hot days, cooler city water is added to lower the temperature of the condensing water to the refrigerating machine. A 3 in. city water line was extended to a 1000 gal open tank, the water level being governed by a ball cock. The suction of a 200 gpm pump is connected to this tank, and this new pump discharges into the suction of the existing circulating pump so that the colder city water is mixed with the warmer water from the cooling tower before it goes through the condenser on the air conditioning refrigeration machine. The overflow goes over the roof to the downspouts. This supplemental city water is used only on the relatively few days of high outdoor wet bulb temperature.

The attention of the operating staff was also drawn to the necessity of thoroughly cleaning out the condenser every winter, and inspecting all spray nozzles so that the equipment is in proper condition to function effectively each spring.

The writer served in an advisory capacity in connection with these changes.

*Chief Engineer, Chicago Board of Education. Member of Board of Consulting and Contributing Editors.

Heat Barriers Classified

In his discussion of types and applications of building insulation at the University of Illinois air conditioning conference last month, S. Konzo, special research assistant professor of mechanical engineering at the University, classified arbitrarily heat barriers according to the following scale: Types of wall construction which have a U value (Btu per hr per sq ft per deg of temperature difference) of 0.10 or under were classified as excellent; those with a value between 0.10 and 0.20 as good; those from 0.20 to 0.30 as fair; and those with a U value of over 0.30 as poor.

Fabrication Details for High Pressure and Temperature Piping at Delray

By Arthur McCutchan*

ATTACHMENT of bypass and drip connections, design of thermocouple wells for high temperature service, improved types of backing rings, standardization of welding bevels, and the appearance of finished welds were discussed last month. In this second article, the reasons are given for changing the method of beveling welded branch connections to permit the use of backing rings, and the re-design of the ring type reinforcement to give greater support at the sides of welded branch connections is described.

Reduced Outlet Welded Branch Connections

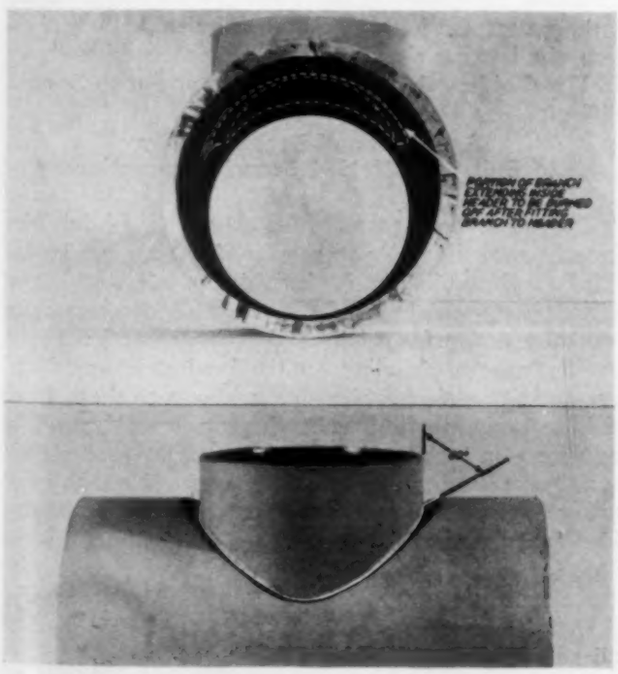
The type of reduced outlet welded branch connection shown in Fig. 1 was used with the first 75,000 kw unit of the Delray extension. This type was not wholly satisfactory in that an excessive amount of cutting and beveling was required on both parts in fitting them together, which made the operation difficult to control by templates. Furthermore, on trial it was found impracticable to trim off the dotted portion (see Fig. 1) of the branch which extended within the header, since doing so produced a feather edge against which it was difficult to start the weld. Hence with this type it was found expedient to leave the branch projecting inside the header as shown despite the disturbance to flow conditions. In common with all welded pipe joints not using a backing ring, the completeness of penetration to the inside of the pipe wall was a matter of conjecture.

In order to simplify preparation of the joint and at the same time provide more satisfactory conditions for welding, the type of reduced outlet welded branch connection shown in Fig. 2 was developed. In this design a backing ring with spacing pins has been applied to the type of beveling for welded branch connections popularized by F. C. Fantz.[1] The resulting design possesses obvious advantages from the standpoint of simplicity in cutting and fitting the mating parts, plus the advantage of using a backing ring with spacing pins which facilitates penetrating to the bottom of the groove, thus assuring that the designed weld cross-section will be secured at all points around the intersection.

The sketches of Fig. 3 show the shape of backing ring necessary to fit the contour of the intersection of a reduced size branch and header.

*Engineer, Engineering Div., The Detroit Edison Co.

Fig. 1—Method of beveling reduced size welded branch connections—header only beveled



Full Size Welded Branch Connection

In investigating stress conditions in a full size welded branch connection,[2] high stresses were found to exist on the *outer* surface at the flat portion on the sides of the manifold. In the vicinity of the crotch, high stresses were deduced analytically as existing in the *inner* surface, but were not determined experimentally. The ring type reinforcement shown in Fig. 4 was found to adequately reinforce the crotch but did not prevent distortion under pressure at the sides of the manifold to the extent desired. Although the pressure at which failure occurred was approximately the same as would be expected to burst a straight length of pipe, the location of the failure, just below the reinforcing ring, was not as

[1] "Design and Fabrication of High Pressure, High Temperature Welded Piping," by F. C. Fantz, abstracted in HEATING, PIPING AND AIR CONDITIONING, May, 1938, pp. 329-332.

[2] "Investigation of Stress Conditions in a Full Size Welded Branch Connection," by F. L. Everett and Arthur McCutchan, ASME *Transactions*, FSP 60-12, July, 1938, pp. 399-410, and discussion February, 1939, pp. 167-176.

Carbon-molybdenum steel is being used for the 865 lb, 910 F steam piping in the extension of Detroit Edison's Delray Station, and in the January and February issues of HEATING, PIPING AND AIR CONDITIONING, Sabin Crocker reported on experience in welding this piping material. . . . Last month, Arthur McCutchan discussed fabrication details adopted for this piping, covering attachment of bypass and drip connections, thermocouple wells for high temperature service, backing rings, welding bevels, and appearance of finished welds. He concludes this series this month with his discussion of the re-design of the ring type reinforcement and the method of beveling welded branch connections to permit contoured backing rings

Fig. 2—Reduced outlet welded branch connection with backing ring

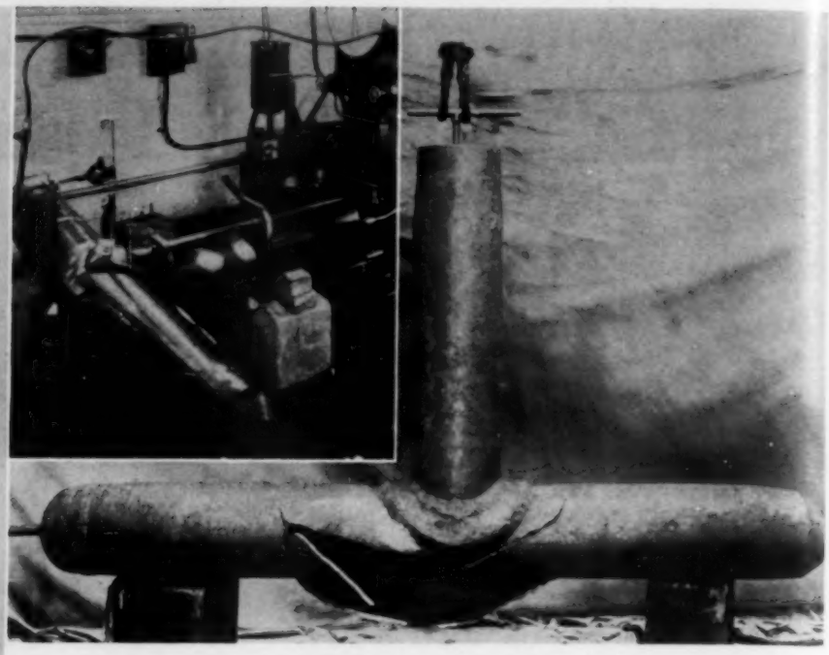


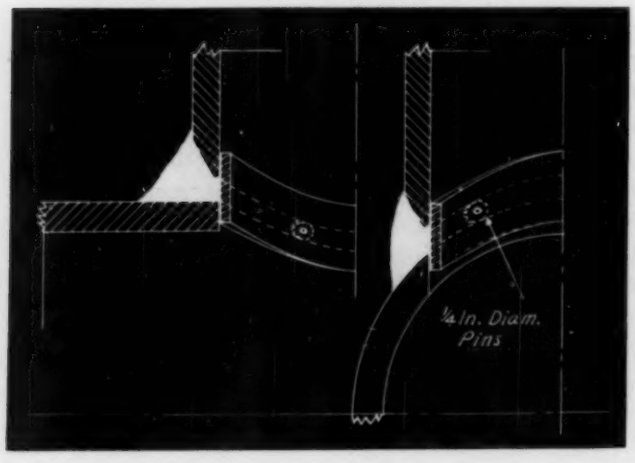
Fig. 4—Welded pipe manifold for 650 lb, 850 F service which burst on hydrostatic test at 8750 lb

reassuring as if failure had occurred in the straight pipe entirely outside of the region affected by making the branch connection.

On completion of the tests, the welded branch connection was carefully explored by thoroughly sectioning the weld and conditions were studied throughout. It was apparent, as may be noted from Fig. 5, that full penetration was not obtained at some points, principally because of lack of proper spacing of the parts to afford sufficient room at the root of the weld. Additional sections cut from the sides of this manifold were shown in the February, 1939, HEATING, PIPING AND AIR CONDITIONING, p. 92. No failures occurred in this welding when subjected to some 1200 loading-unloading cycles during the investigation and in the final test to destruction. Since the strength of the weld metal deposited from low carbon electrodes is less than that of the medium-carbon silicon-killed pipe material used in this manifold, it is evident that the sections provided must have contained sufficient allowances for voids and lack of penetration.

The method of beveling branch and header for a full size welded branch connection of the type used with the first 75,000 kw unit at Delray and for the test manifold is illustrated by the photograph of a wooden model, Fig. 6. The preparation of the branch and header for this type of connection required considerable skill in cutting and fitting. The amount of weld metal required was excessive and it was not possible to separate the parts sufficiently to enable the welder to secure complete penetration to the bottom of the welding vee without getting weld spatter and icicles inside the header. The application of a backing ring to this type of intersection shown in Fig. 6 of branch and header would appear to be quite difficult.

Fig. 3—Sketches showing shape of backing ring to fit contour of intersection of reduced size branch and header



Application of Backing Rings to Full Size Welded Branch Connections

The component parts of another type of full size branch connection to which a backing ring can be applied are shown in Fig. 7. This design is a modification of that suggested by F. C. Fantz.[1] To obtain a more workable design for a full size branch connection and to avoid excessive grinding on the header, it was found necessary to make supplementary V cuts at the sides of the header and to do some beveling on these V cuts. The addition of a backing ring is, of course, the most significant change and an improvement not feasible with other methods of preparation. The branch connection assembled for welding is shown in Fig. 8. The use of a backing ring with this method of preparation permits adequate separation of the branch and header so that the designed weld cross-sections actually can be obtained at all points around the intersection.

Re-Design of Reinforcement

In addition to bringing out the desirability of changing the method of preparing branch connections for welding, investigation of the test manifold pointed the need for some modification of the reinforcing ring to give greater support at the sides of the manifold. The results obtained from strain gage measurements on the manifold before applying the reinforcing ring were especially informative. The heart shaped stress contour lines shown on Fig. 9 indicate that the sides of the manifold have a definite tendency to assume a spherical shape.

Tests on the same manifold with the ring type reinforcement added (see Fig. 4) showed that some permanent distortion occurred at the sides at a pressure only 70 per cent of that calculated to cause yielding of the pipe proper. In re-designing the ring reinforcement for the new style of welded branch connection, the bottom of the ring was extended to cover the region where final failure of the test manifold occurred (see Fig. 4), and side plates have been added to span the highly stressed portion of the branch. The reinforcing ring and one of the side plates are shown in Fig. 10 in position on the welded branch prior to final forming of the bottom ring around the header. Making the side plates separate from the ring simplifies forming both parts of the reinforcement and permits welding the reinforcement to the branch connection at the center of the flat portion, thus affording more firm support than is possible where this area is bridged across. With the reinforcement attached in this manner there is reason to believe that it will assume its full share of the pressure loading. The final appearance of a 10 in. pipe header with the new style reinforcement is shown in Fig. 11. Minor modifications of this design may be found desirable as further experience is had in the field, but it is believed to embody the findings of the investigation of the welded test manifold[2] and the constructive suggestions offered in discussion of that investigation.

Additional tests[3] have confirmed the finding that the method of determining the reinforcement of the crotch by compensating for material removed in cutting the branch opening in accordance with ASME boiler code rules for reinforced nozzle openings affords a satisfactory design basis. This conclusion is of fundamental importance, since the boiler code rules were not intended to apply to openings of a diameter greater than 60 per cent of the shell diameter. The need for providing greater support at the sides has been further em-

[3]Unpublished data, manifolds tested by Crane Co. in co-operation with the subcommittee on welded branch connections of the ASA code for pressure piping.

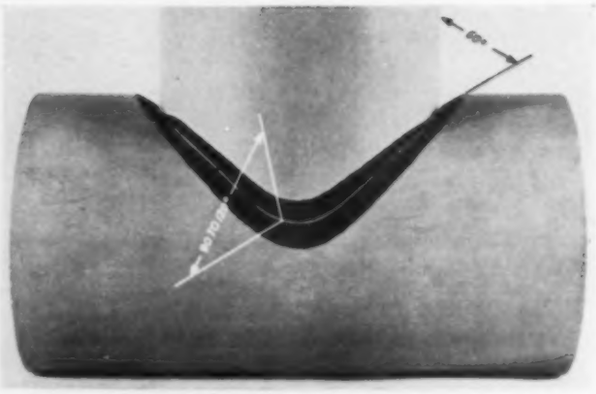


Fig. 6—Model showing method of beveling branch and header—welded test manifold

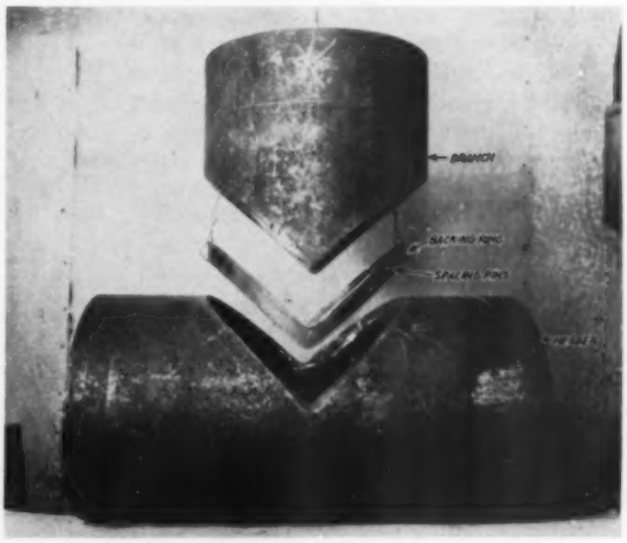


Fig. 7—10x10 in. Schedule 80 pipe branch connection with backing ring showing component parts

Fig. 5—Sections through crotch of welded test manifold

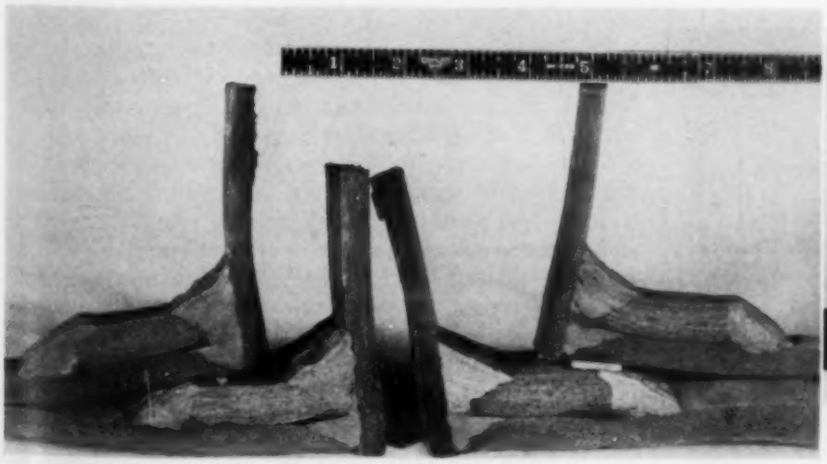


Fig. 8—10x10 in. Schedule 80 pipe branch connection with backing ring assembled for welding

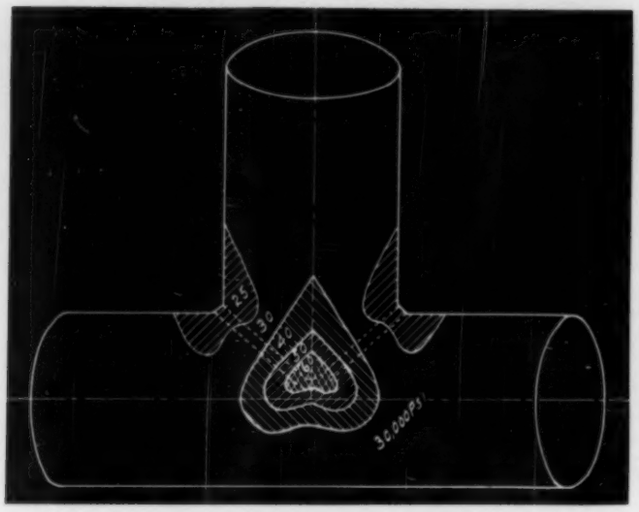


Fig. 9—Stress in outer surface of unreinforced manifold—internal pressure 3412 psi (lb per sq in.)

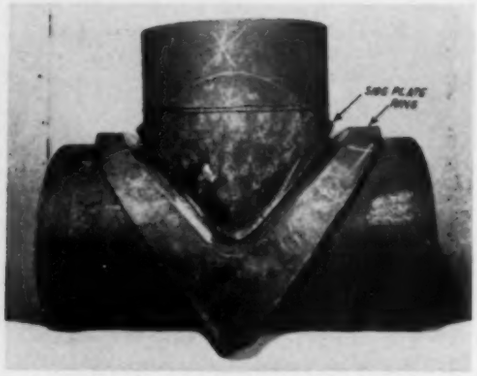


Fig. 10—10x10 in. Schedule 80 pipe branch connection with backing ring showing reinforcing ring and side plate in position prior to final forming of the ring around the header

Fig. 11—10x10 in. Schedule 80 pipe branch connection with backing ring showing completed weld with new style reinforcement

phasized as well as the desirability of extending the reinforcement both lower on the header and higher on the sides of the branch.

This concludes the fourth article of a series describing Detroit Edison practice in welding carbon-moly and other high pressure piping in the current extension of Delray power plant. The present articles supplement several previously published accounts of this company's work in developing satisfactory technique in applying this useful, but rather difficult, art to its high pressure, high temperature pipe joints. These developments represent the combined efforts of the company's engineering, drafting and field forces, working together as described on p. 90 of the February HPAC.

Owing to the comparative newness of the application, there still remains a promising field for pioneer effort in perfecting immature practices and evolving novel details peculiarly suited to pipe welding. Only when the art grows up and a deadening sense of perfection has been achieved, will interest lag in originating new methods and improved construction suited to welding. In the meantime, progress is being furthered through the free exchange of ideas fostered by professional societies, technical magazines, and the trade in general, and the authors of this series of articles are pleased to acknowledge on behalf of their company and themselves full appreciation of their indebtedness to others and the cooperative spirit shown by all.

Reverse Cycle System Heats or Cools Building

Automatically holding a 70 deg temperature, the reverse cycle air conditioning system installed in the new Westinghouse Electric & Mfg. Co. building at Emeryville, Calif., heats or cools, corrects humidity, and circulates 16,000 cfm of air. The estimated cooling load is 196,000 Btu, and the estimated heating requirement is 238,400 Btu with a 35 deg outside temperature. The building is 90 ft long, 50 ft wide, two stories high, with 6 in. concrete walls and roof insulated with 4 in. redwood bark.

The pick-up ratio of the heat pump installation with a 35 deg outdoor temperature and a 70 deg room temperature is approximately five to one. In other words, for every kwhr of electrical energy consumed, the equivalent of approximately 5 kwhr of heat energy is obtained. The plant, which has two 7½ ton, hermetically sealed compressors, was designed for manual or full automatic operation, with necessary thermostats, humidistats, and protective devices to maintain constant temperature, air circulation, air filtration, and humidity control for summer and winter conditioning. A 7½ hp motor operates each compressor, and each fan is operated by a 3 hp motor, making a total connected load of 21 hp. The system is of the central plant type.

Power Savings Through Static Pressure Regain in Air Ducts

THE power required to drive a fan is proportional to the product of the volume of air handled and the total pressure. The total pressure is the sum of the pressure lost at the entrance to the system, the pressure lost from friction and shock[1] throughout the system, and the velocity pressure corresponding to the velocity of the air as it leaves the system. Reducing the velocity of the air before it is released from the system, by causing it to pass through a properly designed enlarged section, will decrease the velocity pressure at the end of the system. The energy represented by this decrease in velocity pressure may be utilized to counterbalance a part of the friction and shock losses of the system, thereby decreasing the total pressure and, in turn, the power required.

—By J. R. Fellows*—

Reducing the velocity of the air before it is released from a duct by causing it to pass through a properly designed enlarged section will decrease the velocity pressure at the end of the system. The energy represented by this decrease in velocity pressure may be utilized to counterbalance a part of the friction and shock losses of the system, thereby decreasing the total pressure and, in turn, the power required. . . . Theoretical and test data are presented here, with the author's recommendations for applying the principle

Professor E. Peclet[2] found, in 1861, that a funnel attached to the end of a straight tube conducting air from a tank in which constant pressure was maintained would increase the velocity of the air in the straight tube, provided the total included angle between the sides of the funnel was not more than 40 deg. Funnels having a greater angle than 40 deg had no effect on the velocity in the straight tube.

Peters,[3] and Kratz and Fellows,[4] found the effectiveness of an expanding section at the end of a system for converting velocity pressure into static pressure is increased by addition of an after section of straight duct (see Fig. 1) the size of the leaving end of the expanding section.

Static Pressure Regain Applied to a Fan and Duct

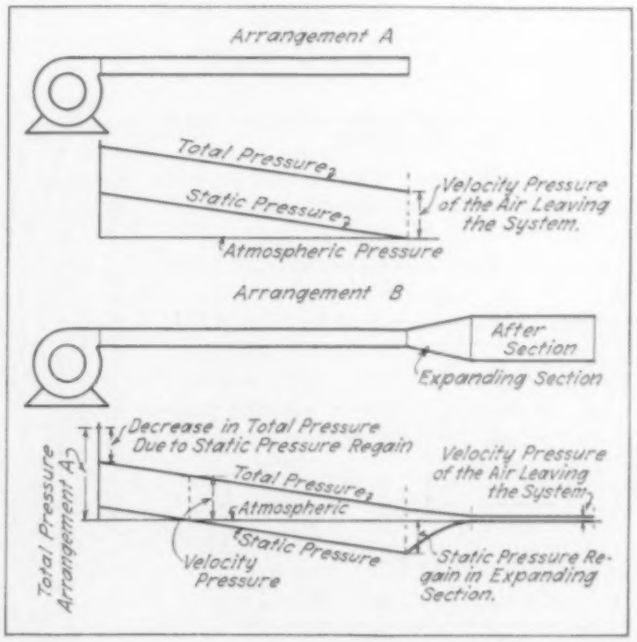
Fig. 1 illustrates use of static pressure regain in a simple system consisting of a fan and discharge duct. The fan in the upper part of the figure has a free inlet and discharges the air through a straight duct having a diameter equal to that of the fan outlet (arrangement A). In this case, the total head or total pressure which must be provided by the fan is the static pressure at the fan outlet or the pressure that is necessary to overcome the friction pressure losses in the duct, plus the velocity pressure which, in this case, is the same at any location along the length of the duct. A similar system with an expanding section and after section added at the leaving end (arrangement B) is shown in the lower part of Fig. 1. In arrangement B the velocity is decreased in the expanding section before the air is released to the atmosphere.

It may be noted that with arrangement B the total pressure at the fan outlet is less than when arrangement A is used. The addition of the expanding section and after section to arrangement A lowers the operating cost of the system and permits the use of a smaller fan motor. The percentage of the power required with arrangement A that may be saved by adding the expanding section and after section is limited by the ratio of the velocity pressure to the total pressure. If the velocity pressure is one-half of the total pressure, as in Fig. 1, nearly one-half of the power requirements of arrangement A may be saved by the application of static pressure regain.

Regain in an Abruptly Expanded Section

An analysis of the flow through an abruptly expanded

Fig. 1—Application of static pressure regain to a simple fan and discharge duct



*Associate in Mechanical Engineering, University of Illinois.

[1]Shock losses are losses of pressure due to bends and area changes.

[2]A résumé of Peclet's experiments with small funnels is given in a paper by Briggs and Williamson entitled Experimental Study of Fan Evasés, published in the *Transactions* of the Institution of Mining Engineers, Vol. 68, p. 323, 1924-25.

[3]Conversion of Energy in Cross-sectional Divergences Under Different Conditions of Inflow, by H. Peters. National Advisory Committee for Aeronautics, Technical Memorandum No. 737. Translated from *Ingenieur-Archiv.*, Vol. II, 1931.

[4]Pressure Losses Resulting from Changes in Cross-sectional Area in Air Ducts, by Alonzo P. Kratz and Julian R. Fellows. *Bulletin* No. 300, Engineering Experiment Station, University of Illinois. 1938. Figs. 2, 3, 4 and 5 of this article are from this bulletin.

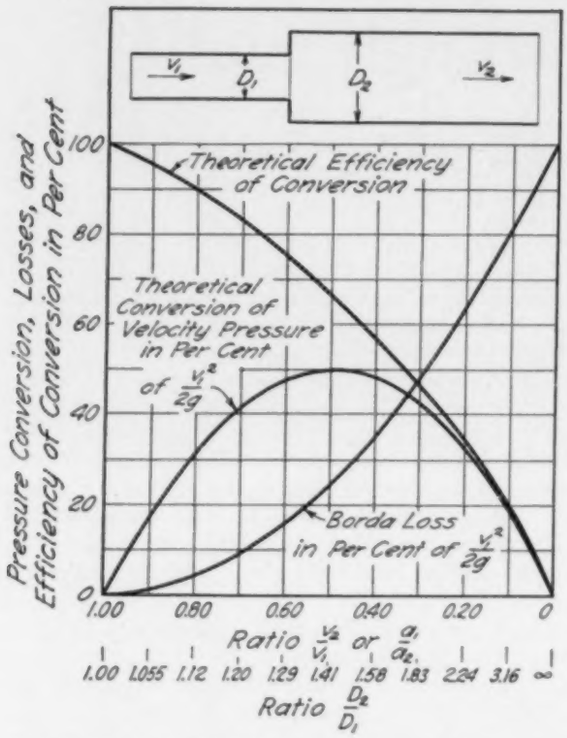

Pressure Conversion, Losses, and Efficiency of Conversion in Per Cent
Theoretical Efficiency of Conversion
Theoretical Conversion of Velocity Pressure in Per Cent of $\frac{V_1^2}{2g}$
Borda Loss in Per Cent of $\frac{V_1^2}{2g}$
Ratio $\frac{V_2}{V_1}$ or $\frac{a_1}{a_2}$
Ratio $\frac{D_2}{D_1}$

Fig. 2—Theoretical velocity pressure conversion, pressure losses, and efficiency for an abrupt expansion

section indicates that such a section, when placed at the end of a system, would effect a static pressure regain.

Referring to Fig. 2, since the velocity V_2 in the larger pipe is less than the velocity V_1 in the smaller pipe, the velocity pressure in the larger pipe would be less than the velocity pressure in the smaller pipe, and if there were no loss of head from shock or friction, the regain in static pressure would be equal to the decrease in velocity pressure, or

$$S_r = \frac{V_1^2}{2g} - \frac{V_2^2}{2g} \quad \text{.....[1]}$$

where S_r = regain in static pressure, feet of fluid flowing;
V_1 and V_2 = mean velocities in the smaller and larger pipes respectively, feet per second;
g = acceleration due to gravity, feet per second per second.

The Borda equation for computing loss of head in an abrupt expansion is

$$H_L = \frac{(V_1 - V_2)^2}{2g} \quad \text{.....[2]}$$

where H_L = pressure or head lost, feet of fluid flowing.

The regain in static pressure would then be the difference in velocity pressures, minus the head lost:

$$S_r = \left[\frac{V_1^2}{2g} - \frac{V_2^2}{2g}\right] - \left[\frac{(V_1 - V_2)^2}{2g}\right] \quad \text{...[3]}$$

Equation 3 simplifies to

$$S_r = \frac{V_1 V_2 - V_2^2}{g} = \frac{V_2 (V_1 - V_2)}{g} \quad \text{.....[4]}$$

The efficiency of an abrupt expansion in converting a decrease in velocity pressure into a static pressure regain may be defined as the ratio of the regained static pressure to the change in velocity pressure, or

$$\frac{\frac{V_2 (V_1 - V_2)}{g}}{\frac{V_1^2 - V_2^2}{2g}} = \frac{2V_2}{V_1 + V_2} \quad \text{.....[5]}$$

The theoretical conversion of velocity pressure to static pressure regain in an abrupt expansion as given by Equation 4 expressed in per cent of $V_1^2/2g$ is plotted in Fig. 2 and shows that, theoretically, the maximum static pressure regain occurs in an abrupt expansion when the after section has a cross-sectional area two times the cross-sectional area of the approach section. The theoretical efficiency as given by Equation 5 and the head lost as given by Equation 2 also expressed in per cent of $V_1^2/2g$ are likewise plotted in Fig. 2.

The results of Archer's[5] experiments with water flowing from a brass pipe to one of a larger cross-sectional area, including a wide range of area ratios, confirm the foregoing theory and the theoretical curves of Fig. 2.

Regain in a Gradually Expanded Section

Experiments by Gibson,[6] Peters,[3], Briggs and Williamson,[2] and Kratz and Fellows,[4] show that the loss of head in tapered transitions with total included angles of less than 40 deg is less than the Borda loss for abrupt expansion given by Equation 2.

The loss in such tapered transitions may be represented by

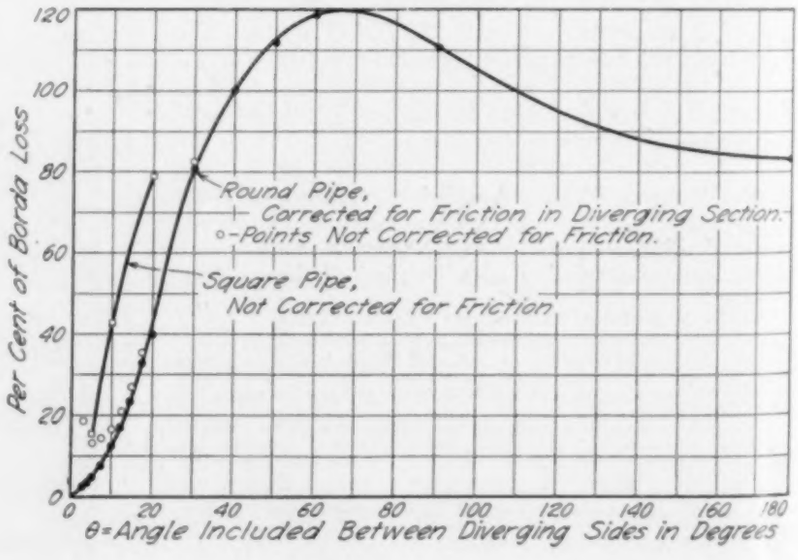
$$\frac{K (V_1 - V_2)^2}{2g} \quad \text{.....[6]}$$

where K is the ratio of the loss in a tapered expanding section to

[5]Experimental Determination of Loss of Head Due to Sudden Enlargement in Circular Pipes, by W. H. Archer. *Proceedings*, ASCE, Vol. 39, Part I, p. 365, 1913.

[6]Flow of Water Through Pipes and Passages Having Converging or Diverging Boundaries, by Prof. A. H. Gibson. *Proceedings*, Royal Society of London, Vol. 83, 1910.

Fig. 3—Ratio of actual loss in diverging sections to Borda loss, from data of A. H. Gibson


Per Cent of Borda Loss
Round Pipe, Corrected for Friction in Diverging Section.
o-Points Not Corrected for Friction.
Square Pipe, Not Corrected for Friction
θ=Angle Included Between Diverging Sides in Degrees

the loss in an abruptly expanding section for otherwise identical conditions of flow. Fig. 3 from the data by Gibson gives values of K for angles of divergence to 180 deg.

The static pressure regain in a tapered transition followed by an after section may be expressed directly in terms involving the change in velocity pressure, the Borda loss and the constant K, as follows:

$$S_r = \left[\frac{V_1^2}{2g} - \frac{V_2^2}{2g}\right] - \left[\frac{K\,(V_1 - V_2)^2}{2g}\right] \quad \ldots\ldots [7]$$

Equation 7 may be more conveniently written as

$$S_r = \frac{V_1^2 - V_2^2 - K\,(V_1 - V_2)^2}{2g} \quad \ldots\ldots [8]$$

In the case of air flowing in ducts the static pressure regain expressed in inches of water becomes

$$S_r = \frac{12\,d_a}{d_w}\left[\frac{V_1^2 - V_2^2 - K\,(V_1 - V_2)^2}{2g}\right] \quad \ldots\ldots [9]$$

where d_a = density of air, lb per cu ft;
d_w = density of water, lb per cu ft.

Theoretical curves showing the static pressure regain in tapered transitions followed by after sections, calculated with Equation 8 and using values of K from Fig. 3, are shown in Fig. 4 and give the per cent of the velocity pressure in the approach section that would theoretically be converted to static pressure when different included angles of divergence are used, as well as different ratios of areas between the approach section and after section.

It may be noted from Fig. 4 that when the expanding section has a total included angle less than 10 deg, the after section may advantageously be made to have a cross-sectional area equal to four or more times that of the approach section, and that the per cent of $V_1^2/2g$ that may be converted to static pressure may theoretically be made to approach 100. Because of greater length, the loss of head due to friction with the pipe wall in transitions with small included angles nullifies a part of the greater static pressure regain.

Fig. 5 from data by Kratz and Fellows[4] shows actual static pressure gradients for four tapered transitions connecting a 6 in. by 6 in. duct used as the approach section and a 11¾ in. by 11¾ in. duct used as the after section. The curves shown are actual measured static pressure gradients and the decrease in static pressure below atmospheric pressure at the connecting plate represents actual net static pressure regain effected by the expanding section and after section in each of the arrangements for which a curve appears.

It may be noted that the expanding section with an included angle of 3 deg was more effective in converting velocity pressure to static pressure than any of the others tried. The expanding section with an included angle of 7 deg in the arrangement as shown, effected a static pressure regain that was approximately 94 per cent of that effected by the 3 deg expanding section. The ratio of the area of the after section to that of the approach section was in every case the same and equal to 3.84. The theoretical curves in Fig. 4 indicate that the area ratio of 3.84 is not the optimum one for 30 deg and 60 deg expanding sections. The static pressure regains effected in these sections would have been somewhat greater than the values shown for them in Fig. 5 if the

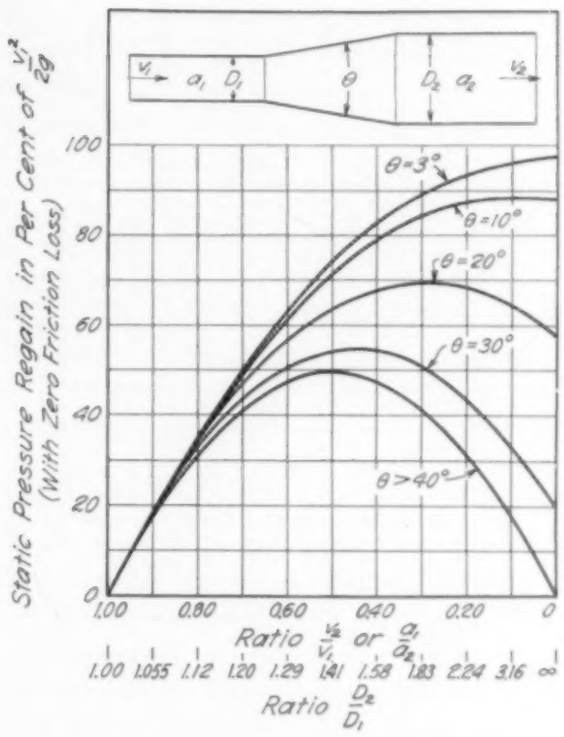


Fig. 4—Static pressure regain for diverging sections in smooth circular ducts

area of the after section had been made approximately two times the area of the approach section.

Two Arrangements for Static Pressure Regain

Fig. 5 indicates that the addition of an after section to an expanding section having a total included angle of 3 deg does not materially increase its effectiveness in converting velocity pressure to static pressure. When an expanding section having a total included angle of 7 deg is used, the expanding section should be followed by an after section having a length equal to not less than six diameters and not more than 14 diameters of the approach section. In either case, the diameter of the expanding section at the leaving end should be approximately two times the diameter at the entering end. Fig. 6 shows suggested dimensions for a 3 deg expanding section and after section (design 1) and a 7 deg expanding section and after section (design 2) given in terms of the diameter of the approach section.

The 3 deg expanding section with after section as shown in design 1 is recommended for effecting static pressure regain at the end of the system where space limitations do not prohibit its use. The after section may be omitted if desired with a sacrifice of not more than 3 per cent of the static pressure regain that would be effected by the complete arrangement.

If space limitations will not permit the use of an expanding section and after section having an over-all length of more than 14 diameters of the approach section of duct, the use of an expanding section having a total included angle of 7 deg followed by an after section having a length equal to six diameters of the approach section (design 2) may be used. The curves in Fig. 5 indicate that the static pressure regain effected by this design would be approximately 6 per cent less

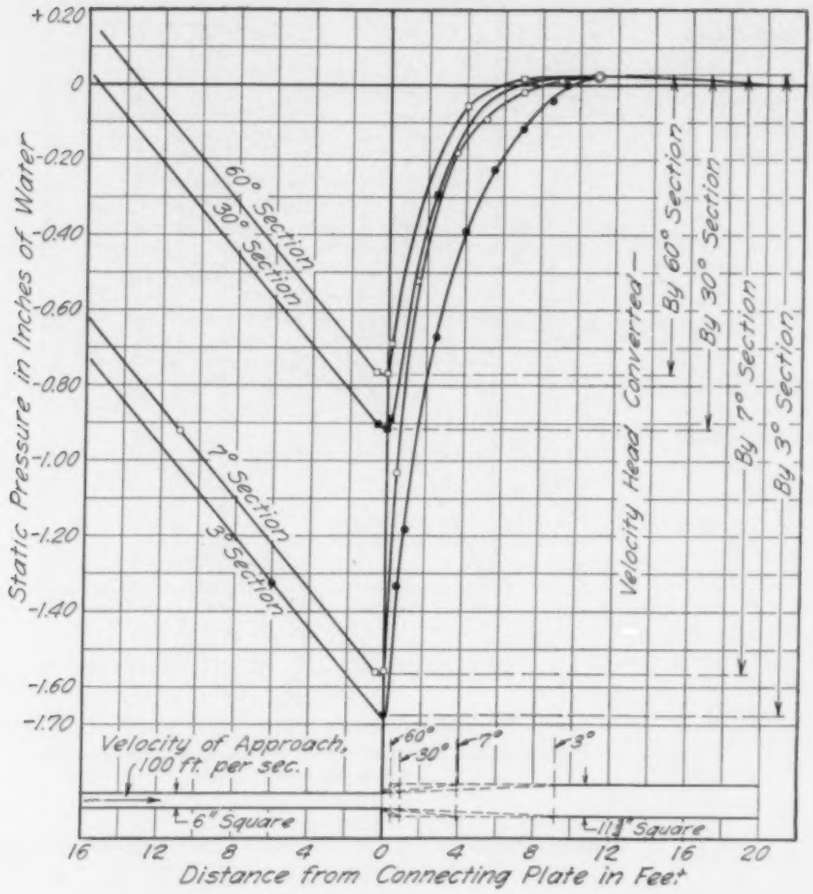


Fig. 5—Observed static pressures in duct with diverging sections

than that effected by design 1. Where the omission of the after section from design 2 is absolutely necessary because of space limitations, the 7 deg expanding section may be used alone, but from Fig. 5 it appears that the static pressure regain will not be more than 85 per cent of that which would be effected by design 1.

More experimental data are required to determine the variation in the efficiency of the proposed designs with variation in air velocity and in size of the approach section. However, from a study of the data taken by Peters,[3] whose test pipe had an approach section 70 mm in diameter, and the data of Kratz and Fellows,[4] whose test pipe had an approach section 6 in. square, the author is of the opinion that the efficiency of the proposed designs will be approximately the same for all pipe sizes and air velocities that are apt to be used commercially.

Designs 1 or 2 as shown in Fig. 6 may be attached at the discharge end of any duct for the purpose of effecting static pressure regain.

Effect on Power Requirements

If the fan speed is left unchanged when the installation of the expanding section and after section is made, the static pressure at the fan outlet will be lower, with the result that the fan will deliver a greater volume of air. The fan power requirements may be increased or decreased, depending on the fan design and its previous loading. If the fan in use has a power characteristic which rises with increased volume delivered, the power required would be increased.

Example—A certain fan with forward tipped blades delivers 16,880 cfm at 311 rpm and requires 6.65 hp to drive it when the static pressure is 1¾ in. of water. If the static pressure is reduced to 1 in. of water and the fan speed left unchanged it will deliver 31,650 cfm and require 15.0 hp. However, if the fan speed were decreased to 235 rpm, the original 16,880 cfm would be delivered against a static pressure of 1 in. of water with an expenditure of 4.23 hp.

If static pressure regain is applied to save power, the amount of the static pressure regain should be calculated by the use of Equation 9 and deducted from the static pressure of the system as previously operated. A fan speed should then be selected from the fan manufacturer's catalog data that will supply the original volume of air against the decreased static pressure. The horsepower requirements of the fan at the new speed and static pressure should also be obtained from the same source. The efficiency of the motor at the new loading should then be determined from a typical efficiency curve for the type of motor being used. If the original motor will not operate efficiently at the new loading it will be necessary to replace it by one that will, if maximum savings are to be effected. A pulley should then be selected that will cause the fan to be driven at the correct

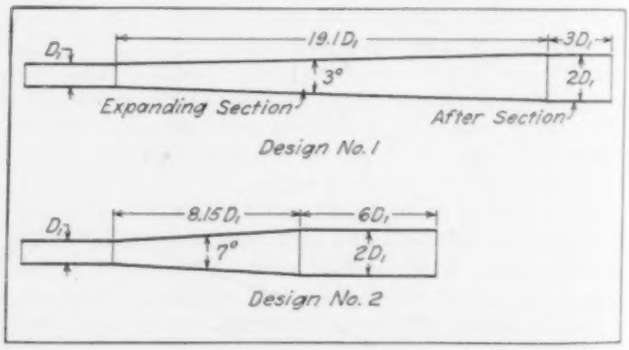


Fig. 6—Expanding sections with after sections for effecting static pressure regain

speed for the new operating conditions. In most practical applications of static pressure regain it would be practicable to use the original motor for driving the fan and it would only be necessary to change the pulley.

Costly losses due to discarding ampules containing minute dust particles accumulated in the packaging of sensitive cobra venom solution and gland extracts have been reduced by two-thirds since the filling room in which this operation is carried out at the Hynson, Westcott and Dunning plant in Baltimore has been air conditioned, Carrier Corp. reports.

Preparing the Piping Estimate

By George E. Deatherage*

To cover the subject of piping estimates and costs completely would require at least a volume. What the author does here is to indicate the importance of a proper procedure in making up the bill of materials and to show how it is used in determining cost . . . By following through a logical scheme, economies in purchasing, erection and other work are made possible

THE subject of piping estimates and costs is difficult to discuss from a practical viewpoint, for the fabrication and erection of piping involve many variables. Costs start with the plans and follow through with the billing of the material, purchasing, delivery to the job, fabrication, erection, and testing. All must be considered in preparing a piping estimate, not forgetting allowances for field conditions.

Properly prepared piping plans are a large factor both in making the estimate and performing the work. There are a few principles that should be followed for best results.

If possible, the scale should not be less than ¼ in. per ft.[1]

Separate plans should be made for steam, water, gas, etc., if putting them all on one sheet will crowd it so as to make it difficult to interpret.

Each line or branch should be given a number on the plan to identify it on the bill of material and shop sketches.

Sufficient elevations should be made in order to make the drawing readily intelligible.

Each draftsman or engineer should use the same terminology and piping symbols.

Particular attention should be given to clearance between lines for insulation, hangers, and other interferences.

There is no advantage in cheapening the plans at the expense of idle time on the job. It is more economical to spend money on a drawing than to waste it in lost time and errors in the field.

Billing the Material

The first step in preparing the estimate is listing the materials required. This is best done on a regular form for this purpose, accurately enough so that if the job goes ahead the estimate can be used to purchase the material without having to do the task over again. A form for this purpose is illustrated here; it is used as follows:

Bill of Material Number—Each sheet of the bill is numbered with the number of the general drawing covered, as well as by the number of the line or branch. This identifies the job and the specific line to which the list applies. Only one line or branch should appear on each sheet, and all of the material for it listed, including bolts, gaskets, hangers and all specialties. With only one line or branch to a sheet, the listing lends itself to economies as the work proceeds and in the estimating. Each sheet can be priced separately. If desired a percentage of the total may be taken for field labor, or number of man-hours required for the fabrication and erection of each line may be determined; this will be discussed more fully later.

The foreman in the field can be given a copy of the material list so that he knows exactly what has been ordered for each line of the work. In practice, the author has supplied the stores-keeper with a copy of the lists, by which he checks the incoming material, bags the fittings and delivers the complete material for any line or group of lines to the job site, at the request of the foreman.

Drawing Number—The number of the general drawing covering the piping is carried by itself in the upper left-hand corner for purposes of cross-reference. Revisions are indicated in the upper right-hand corner.

Job—In this space a word description of the job is carried, such as boiler house piping, turbine exhaust, etc.

Contract Number—This space is provided for accounting and filing purposes.

Date Listed—This bears the date on which the material was listed, followed by the name or initials of the person performing the work. Under the item *Charge*, the cost code numbers are indicated.

First step is the listing of materials on a form for the purpose

Dwg. No. _____
Line No. _____ BILL OF PIPING MATERIAL
Sketch No. _____

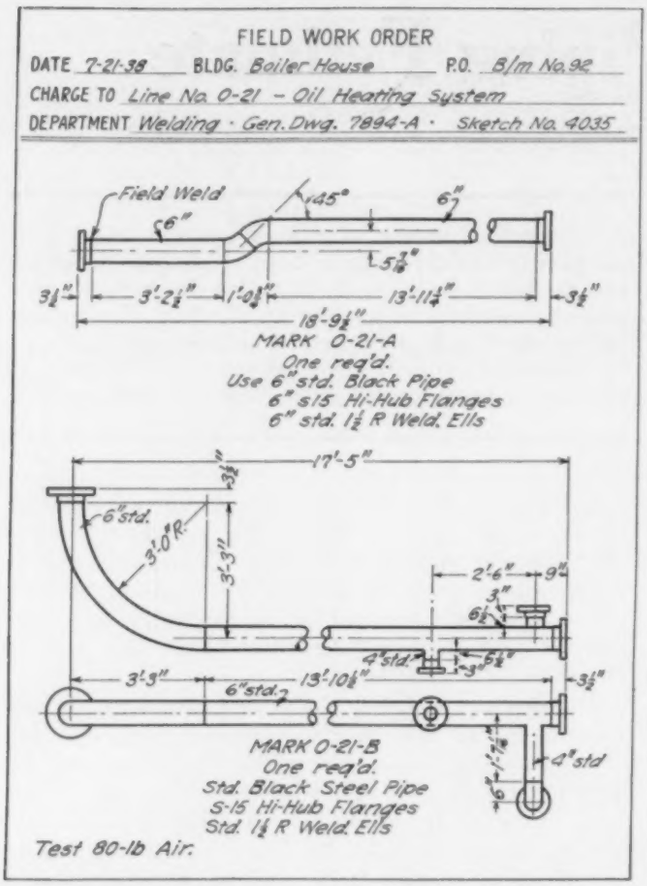
Revision Date	Issue No.

Bill of Material Number _____ Sheet No. _____ of _____
Contract No. _____ Job _____ Date Listed _____ By _____
Job Detail _____ Charge _____
Date to Pur. Dept. _____ Checked _____ Approved _____

No. pcs.	Size	Material - Description	Mark	List Price	Discounts	Net Price	Total Price	Req.	P.O.	Remarks

*Construction Engineer, Mr. Deatherage has been in charge of design, estimating and construction of mechanical installations over a period of some 25 years, most of his experience being with piping work.

[1]The Heating, Piping and Air Conditioning Contractors National Association recently adopted a resolution which states that as the furnishing of small scale drawings and specifications and insufficient information necessary to prepare an estimate intelligently has caused serious loss, it is recommended that no plans (except block plans) shall be drawn to a scale less than ⅛ in. to the foot, and that when ⅛ in. scale plans fail adequately to show or describe the work, larger scale drawings shall be provided.



Shop sketches, fully dimensioned and with essential data for fabrication, are made on letter size sheets

It is important to keep the bills of material in line with any revisions. If the drawing is changed, requiring a change in the type or the amount of materials, the list is revised accordingly. Inasmuch as each list covers only one line, changes in the materials are easily isolated and recorded, so that if necessary they may be sent back for credit or applied to other parts of the work.

Material Listing—The body of the form is self-explanatory, inasmuch as it covers the usual columns for the number of pieces, size, description, identification mark, requisition number, purchase order number, discount, net price, etc. Obviously, all of these columns are not used if the work is not gone ahead with for any reason. However, if the material is properly listed for the estimate, this work needs to be done only once.

Accuracy is the keynote of the whole procedure, and if it is kept in mind that if the work is secured the list is to be used for purchasing it will eliminate the tendency to guess at the number and sizes and automatically result in an accurate estimate.

Pricing the Material Lists

Inasmuch as the material has been listed for purchasing, using a complete description with trade practice terminology as appears in trade catalogs, the work of pricing is facilitated. Assuming that all of the materials that will be required have been listed and have been summarized as to size, number and description, we are in position to achieve economies due to quantity purchasing. With the increased use of special piping materials, economies of this kind are particularly worthwhile.

Freight and Handling—Freight and handling on the job are items that must be paid and as such must be given due consideration. Whether or not these items are priced along with the material will depend upon the method used in figuring field labor. Some estimators calculate the freight as a direct material charge, along with any local cartage. Others, using the percentage method of labor estimating, use catalog prices and include handling in the labor.

It has always seemed to the author that freight and cartage to the job is properly a material charge and that field labor should not start until the material lands on the job.

Estimating Fabrication and Erection Costs

Listing and pricing the materials on the job is purely a mechanical function if performed according to a logical procedure. Itemizing fabrication and erection costs is an entirely different matter. On the larger work, the labor on any one class of work, will be (to some extent) a certain percentage of the cost of the materials, although this percentage varies widely depending on the class of work. In analyzing more than 300 separate jobs, the author found the ratio of labor to material cost varied between 30 and 250 per cent. This is too wide a spread to be worth anything, unless the estimator is thoroughly experienced and knows from his experience what ratio to use.

The percentage method can be used successfully if the estimator has had sufficient experience with it to enable him to set the proper ratios. The cold figures, however, do not describe the conditions under which the work was performed. It has been the author's practice, therefore, never to use the ratio of labor to material if there was a chance to break the work down to definite classifications and calculate from past experience the actual man-hours required. If the materials have been taken off from the numbered lines on the drawing, and each line or branch appears separately, the man-hours required for that specific job may be estimated. The likelihood of error through lack of experience and cost records is still present, but the limits of the error are less than when a single percentage for the entire work is used.

Shop Sketches

Assuming the general piping plan for a welded piping system being well prepared to proper scale, etc., and with each line or branch numbered for identification, each line or branch is broken up into smaller sections for prefabrication in the shop. These sections naturally must be of such a size and shape as will permit handling and erection in the field.

Before the section of piping can be fabricated, a sketch should be made of it for shop use. Whether the shop makes it, or it is made in the drawing room in which the general piping drawings are prepared, is immaterial. The drawing room is the most logical place to make it, and by the same draftsman that made the general piping plan. He is familiar with the work and can more readily make and check the sketches against the plan. This practice lends itself to economy, accuracy and speed.

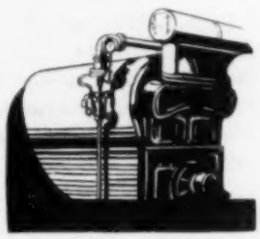
A typical shop sketch (made on letter size sheets for convenience in handling, and to ½ in. scale, fully dimensioned, and carrying essential data for fabrication) is illustrated here. Each section of the sketch should be numbered to key in with the piping plan, for convenience in erection, and should bear the line or branch number. If the work from sketch to fabrication has been accurately performed, the field work is simply that of assembly. As the bill of material number is also carried on the sketch, cross checking is facilitated. Symbols and terminology used by the draftsmen should be standardized so that there will be no confusion.

How Are These Combustion Problems Solved?

[Kalman Steiner* answers a reader's questions]

A READER of HPAC asks about the effect of the amount of moisture in coal on the water vapor in the resulting gases, and the amount of heat liberated per pound of carbon in complete combustion to CO_2 and in combustion to CO.

In reply, the first question can best be answered by making a few typical calculations. In order to simplify the calculations as much as possible, a number of assumptions, will be made which, while not necessarily true, will not affect the usefulness of the solution. Assume, therefore, a coal analysis of moisture 16 per cent, ash 10 per cent, carbon 68 per cent, hydrogen 6 per cent. While of course other substances, such as sulphur, nitrogen and oxygen, would also be present in a stoker fuel, they can be ignored for this calculation.



Combustion calculations are most easily made on the basis of *mols*. A mol is the quantity of a substance whose weight in pounds is the same as the molecular weight of the substance, regardless of the temperature or volume. The molecular weights of interest here are water, H_2O, 18; carbon, C, 12; hydrogen, H_2, 2.

The first step is to express the coal constituents in percentages on an ash free basis. This is done by dividing each percentage of the original analysis by 0.9 (*i.e.*, 1 — 10 per cent ash) to get H_2O, 18 per cent; C, 75.4 per cent; and H_2, 6.6 per cent.

Percentages of constituents are then expressed in mols by dividing each by its respective molecular weight:

$H_2O = 18/18 = 1.0$ mol
$C = 75.4/12 = 6.3$ mols
$H_2 = 6.6/2 = 3.3$ mols

The well known equations of combustion establish the ratios of gas volumes:

$C + O_2 = CO_2$† and $H_2 + 1/2\ O_2 = H_2O$

meaning that each mol of C requires one mol of O_2 for complete combustion and that each mol of H_2 requires one-half mol of O_2. Substituting the mols of C and H in accordance with the above, with the equivalent amount of O_2 for each, we secure the mols of O_2 required.

$6.3\ C + 6.30\ O_2 = 6.3$ mols CO_2
$3.3\ H_2 + 1.65\ O_2 = 3.3$ mols H_2O

Adding: $7.95\ O_2$

Since air is roughly 20 per cent oxygen, there will be four volumes of nitrogen remaining in the flue gases for each volume of oxygen entering combustion. In addition, since the total air required for combustion is $7.95 \times 5 = 39.75$, and since 50 per cent excess air is provided, there will be 19.88 (say 19.9) mols of air in the flue gas (50 per cent of 39.75 = 19.88).

Total Gases in Flue Gas

CO_2 from combustion	6.3
H_2O from combustion	3.3
N_2 from air of combustion	31.8
H_2O from moisture of coal	1.0
N_2 and O_2 from excess air	19.9
Total mols flue gas	62.3

Water Vapor in Flue Gas

H_2O from combustion	3.3
H_2O from moisture in coal	1.0
Total mols H_2O	4.3

Therefore, the percentage of water vapor in the flue gas is $4.3 \times 100/62.3 = 6.9$ per cent.

By a similar calculation it can be shown that if the original coal contained only 3 per cent moisture, the percentage of water vapor in the flue gases would then be 4.5 per cent.

The second question may be answered by taking data on heating values directly from tables. It happens that the heating value of a fuel depends upon the combustion conditions: temperatures of fuel and air at moment of ignition, and temperatures of products of combustion when the released Btu's are measured. Various values will be found in the literature, some based upon 60 deg, others on 32 deg, etc. As a result, for instance, the heating value of CO ranges from 3900 to 4500.

The heat released by burning carbon to CO cannot be measured directly. It is taken as the difference between complete combustion of carbon to CO_2 and the combustion of CO to CO_2. The American Gas Association has published the following data based on temperatures of 32 deg:

Carbon burned to CO_2 yields	169,686 Btu per mol of carbon
Carbon monoxide burned to CO_2 yields	122,328 Btu per mol of carbon
Carbon burned to CO, the difference.	47,358 Btu per mol of carbon

Since the molecular weight of carbon is 12

$169,686/12 = 14,140$ Btu per lb of carbon in complete combustion to CO_2.

$47,358/12 = 3946$ Btu per lb of carbon in combustion to CO.

*Ace Engineering Co.

†The formula for carbon dioxide, CO_2, means that each mol of it contains one mol of O_2 and one mol of C. If carbon could exist as a gas, then a cubic foot of it would combine with a cubic foot of oxygen to make a cubic foot of carbon dioxide, all volumes being at the same temperature. In other words, there would be a shrinkage of 50 per cent in the combined volumes. Carbon of course is not volatilized even at firebox temperatures, so that in effect the equation refers to the amount of solid carbon which would form the required volume if it were gasified. Another way of expressing the reaction quantitatively is to say that 12 lb of carbon combine with 32 lb of oxygen to form 44 lb of carbon dioxide, but there is then no explanation of the volume relations, which are the subject of the original question.

Heating and Ventilation on Treasure Island

THE Golden Gate International Exposition, which opened a few weeks ago on Treasure Island in San Francisco Bay, does not employ air conditioning and heating to the extent that the New York World's Fair will, for the local weather conditions are such that these services are not required in the various exhibit palaces.[1]

The administration building, which is a permanent concrete structure, is heated by means of mechanically circulated hot water, with a boiler plant in the basement consisting of a welded steel boiler fired with oil. The system is operated at a temperature of 215 F, under pressure, with the expansion tank in the boiler room. Radiation is partly the convector type, with direct radiation in some of the office rooms and unit heaters in the large basement areas. Mechanical exhaust ventilation is provided for the main dining room, cafeteria, kitchens, garage, and toilet rooms, as well as for the hospital unit in the building. This building is to become the air terminal building after the close of the exposition, and will be used by Pan American Airways.

The only other permanent buildings are the two hangars, one now used entirely by Pan American; the other, with additions of exhibit rooms on three sides, is the palace of fine arts. The concrete rooms added to the hangar are ventilated by supply fans drawing air from the upper portion of the main hangar, and discharging through diffusing outlets above a wire ceiling in the rooms. No heat is provided, but heat from the lighting is sufficient to keep the building comfortable in the coldest local weather.

No heat or ventilation has been provided in the exhibition palaces except in the palace of foods and beverages, which has a mechanical exhaust system handling 72,000 cfm, part of which is handled through a duct below the floor to which may be connected equipment exhausts. The main exhaust is through the roof with fans and ducts above the roof, but concealed behind the parapet walls.

There are four small auditoriums in the several palaces, each provided with mechanical exhaust at ceilings with air admitted from the main palaces. Festival hall, constructed in one of the palaces, is heated with four circulating unit gas furnace heaters, each of 200,000 Btu input and about 4000 cfm fan capacity. All public toilets in eight locations in palace buildings and three locations in the "Gayway" are ventilated with mechanical exhaust systems.

The police and fire station is heated by a hot water system, gas fired, with direct radiation and unit heaters; there is exhaust ventilation for toilets, lockers, etc. Offices for the housing of various divisions of the exposition, which are situated in the palace buildings, are heated with mechanical warm air gas furnaces. A certain amount of heating of various types has been installed in isolated buildings provided by exhibitors, states, etc.

Fuel for all heating and cooking, with the exception of the administration building, is natural gas brought across the bay from Oakland in a 4 in. submarine line under 50 to 60 lb pressure and reduced at the point where it comes onto Treasure Island to 10 lb. It is then distributed throughout the grounds and again reduced at each consumer's meter.

An interesting installation is the water curtain in connection with the "Cavalcade of the Golden West," a pageant of California history produced in an open air arena with a 200 ft stage. The curtain is provided by a 6 in. header 180 ft long set just below stage level, and containing 2160 vertical nozzles $\frac{1}{32}$ in. in diameter throwing vertical jets of water 25 ft high. Red, blue, and amber flood lights are set immediately behind the header and pointing upward. When water is turned on, a continuous curtain, lighted in colors, obscures the stage for scene changes.

The water is handled with a centrifugal pump in continuous operation; a quick opening gate valve in the discharge line controls the curtain. The waste water runs into two ponds at the front of the stage and is reused. The pump capacity is 350 gpm under 50 lb pressure.

By **William E. Leland***

Twenty million cubic yards of sand were dredged and deposited upon a shoal in the center of San Francisco Bay to make Treasure Island, the mile long, 400 acre site of the Golden Gate International Exposition. After the exposition, the temporary buildings will be removed, runways will be constructed, and the island will become a vast airport. Three of the buildings are of permanent construction, and are to be retained . . . California climate does not require as extensive provisions for air conditioning and heating the exhibit buildings as those at the New York World's Fair described in February. But to keep the record complete, the chief of the mechanical engineering division of the exposition describes the heating and ventilation

*Chief, Division of Mechanical Engineering, Golden Gate International Exposition.

[1]Air conditioning, heating, and ventilating at the New York World's Fair was described in the February HEATING, PIPING AND AIR CONDITIONING by Sanford R. Apt.

What to Consider When Using Wells

AN understanding of the fundamental principles of hydraulics, construction methods and materials used in constructing wells will often prevent the wasting of large sums of money.

Wells are of two general classes, shallow and deep wells. Both classes are constructed in the upper unconsolidated formations, shallow wells being those pumped with surface or suction type pumps and generally less than 100 ft in depth. Deep wells are those over 100 ft deep which are pumped with deep well type pumps. Wells drilled in rock formations are known as deep rock wells. Wells of this type generally range from 200 ft to as much as 2000 ft in depth and are pumped by deep well turbines, displacement pumps or air lift, depending on the well conditions.

The source of ground water is precipitation from the atmosphere. Replenished annually in varying amounts, ground water fills the voids, interstices, cracks, fissures and openings of all kinds below the water table, the upper surface of which is known as the static level. This point is important and should be accurately determined, as it, combined with the drawdown or amount a water bed will lower under a given rate of pumping, determines the number of feet of lift below the surface—all of which determine the cost of getting water from a well to the surface.

Effect of Well Diameter

The diameter of the well is important as it determines the cost and to some extent the amount of water that will be obtained. However, it should be borne in mind that the capacity of a well does not increase directly with the diameter. The capacity depends almost entirely on the yielding ability of the formation in which it is constructed and the methods used in constructing the well itself. All things being equal, a 12 in. well in a sand and gravel formation will yield about 10 to 15 per cent more water than a 6 in. well, while a 48 in. well will produce 20 to 35 per cent more than a 12 in. well and a 10 ft well would produce about 25 per cent more than a 48 in. well. The diameter should always be large enough to accommodate the type of pump to be used on the well, of course.

Well Screens

An important part of any well in a sand-gravel formation is the type and material of the screen. Deep rock wells do not ordinarily use screens, being cased to the rock and obtaining their supply from crevices, fissures and solution channels in the rock formation. The life of screened wells, however, depends almost entirely on the screen used. This is because ground waters are usually either corrosive or incrusting in their chemical nature and immediately start to destroy either the screen itself by corrosion, or its efficiency by building up a hard mineral deposit on and around the screen. These phenomena are well understood by reliable well contractors, who point out these dangers, and the careful executive or engineer will insist on a sieve analysis of the formation to determine the proper size opening to use in the screen, and a chemical analysis of the water in order that the screen may be constructed of a corrosion-resisting metal best adapted for the particular water in hand. So many factors enter into the selection of a well screen that it is practically impossible properly to select a screen without the aid of such analyses.

The capacity of wells is sometimes increased by the introduction of gravel around the screens. This is done by several different methods, probably the most satisfactory being the stabilized gravel pack, where gravel properly graded is introduced around the screen during construction and then followed by proper development. Development is for the purpose of removing the fine sand in the formation surrounding the well, which when done will allow the use of larger openings in the screen.

By E. W. Bennison*

Water is one of the essential materials used in air conditioning, and much of it comes from wells. In localities where it may be had at a low enough temperature, the water may be used for direct cooling of the air. In other cases, well water is used for condensing purposes. In some situations, it is necessary to have a disposal well to get rid of the water after it has served its air conditioning or condensing purposes. . . . . The art of well drilling is of considerable interest to air conditioning engineers and to plant engineers concerned with industrial water supplies. This article tells some of the fundamentals of wells that engineers and executives should remember

Drilling a well to furnish water for air conditioning a department store

*Office Engineer, Edward E. Johnson, Inc.

How a Well Functions

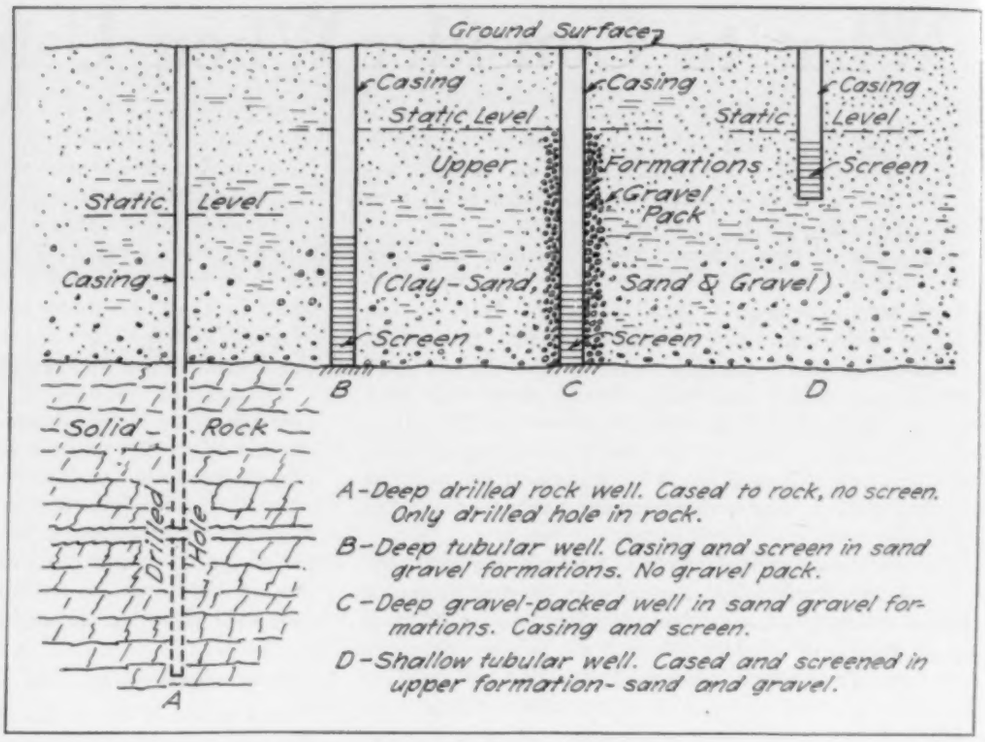
After a well is constructed it is useful to have a general understanding of the way a well functions. The capacity will depend on:

The ability of the formation to yield water according to the depth the well is constructed into it. This relationship is practically that of direct proportion.

The head available or the depth from static water level to the bottom of the well.

The friction loss of the water in motion or the resistance to passage offered by the formation. This varies as the square of the size of the particles making up the formation and in turn controls velocity.

As soon as pumping is started, the water in the well starts lowering and a difference in level between the water in the well and the water in the formation is created. This difference is called drawdown, and is what causes water to move toward the well. The amount of drawdown can be reduced to a minimum by using the proper design and length of screen in the construction of the well. A definite relationship exists between drawdown and yield; for artesian wells this relationship is almost a straight line or direct proportion. For wells in sand-gravel formations the relationship is more of a curve, as indicated by the following:



Sketch illustrating different types of wells

Per Cent of Lowering (Drawdown)	Per Cent of Capacity (Yield)
10	18
20	35
30	51
40	64
50	76
60	85
70	91
80	96
90	99
100	100

The drawdown causes a depression in the water table which extends for a considerable distance around the well depending on the amount of drawdown. For wells of low yield this cone of depression is generally not more than 100 to 500 ft in radius. For larger wells, say 1000 gpm and upwards, it oftentimes extends as much as 1000 to 2000 ft. In other words, the pumping of a well lowers the water table all around for considerable distance and if several wells are situated too closely in the same field, they will interfere and rob each other. In an artesian well this is not the case as the reduction is in pressure rather than levels.

It might be well to repeat that an artesian well is one where the head will rise above the surrounding water table. Some artesian wells flow and some do not. All other wells are non-artesian. The word "artesian" is

[Concluded on p. 239]

Installing a well screen made of alloy material at a brewery

Fig. 1—View of the outdoor sections of the steam and condensate piping

Supporting and Erecting a 24 Inch Steam Tie Line

M. A. Guigou* and S. W. Spielvogel† describe the support and erection of a 24 in. steam tie line and condensate return line connecting two of Consolidated Edison's stations, which are about 500 ft apart. . . . . As explained in the discussion, the requirements were met by utilizing overhead support by means of two suspension cables and with two structural steel towers

THE Consolidated Edison Co. of New York recently completed and put in service a steam line and a condensate return line between its Hudson Ave. and Gold St. stations, which are about 500 ft apart. Purpose of the piping is to convey steam from Hudson Ave. to the turbines at Gold St. for the generation of approximately 100,000 kw. The condensed steam from Gold St. is returned through the condensate line to the boiler feed system of the Hudson Ave. station. The steam line is 24 in. in diameter, has a total length of about 1100 ft, and an ultimate capacity of 1,750,000 lb per hr of steam. The condensate line is 10 in. in diameter.

The support and erection of the outdoor piping is of particular interest and is our subject here; a second article will describe other details, such as the piping joints and materials, pressure reducing and desuperheating equipment, insulation, etc. As will be noted from Figs. 1 and 2, the piping is supported on catenary cables between the two buildings, and there are two intermediate structural towers.

General Features of Piping Design

Design of the 24 in. steam line was controlled by the necessity for protecting the existing steam arteries and various incidental equipment from additional forces superimposed by the expansion of the new piping. In addition it was necessary to confine the new indoor piping to a narrow strip along the building walls of both stations in order to meet clearance requirements for coal lorries, traveling crane and equipment removal space. The piping was fitted in the available space with sufficient clearance for its own elongation and sectionalized by means of anchors into independent units each capable of absorbing its own expansion (Fig. 2).

There is one anchor between Sections 1 and 2, one anchor at Tower B, one at the inlet to Gold St. station, Fig. 3, and a guide at the end of Section 4. The anchors were designed to offer complete restraint for end moments and resistance to pipe thrusts in any direction. The guide was designed to resist thrusts in the horizontal plane but to grant freedom for the vertical expansion of the 16 in. lead to the existing header.

Special provision had to be made to protect an existing valve situated at the foot of this lead. An inspection of the conditions at this point, Fig. 2, will show that hori-

*Division Engineer, Consolidated Edison Co. of New York, Inc.
†Assistant Division Engineer, Consolidated Edison Co. of New York, Inc.

zontal rotations of the 24 in. pipe are translated into a torque in the riser, which in turn is transferred to the header through an intermediate valve. Calculations showed high shearing stresses in the flange bolts of the valve. In order to avoid possible distortions of the valve mechanism the torque was eliminated ahead of the valve by transferring it to the turbine room floor steel through special lugs welded to the riser.

The rate of thermal expansion was taken as 4.9 in. per 100 ft and the modulus of elasticity as 25,500,000 lb per sq in.

Support of Outdoor Piping

Considerable thought was given to the method of supporting the outdoor portion of the line. In connecting the two generating stations the pipe had to cross two city streets and the yard of the Atlantic plant of the National Lead Co., a total distance of almost 500 ft which is encumbered with existing structures and restricted against new structures. An underground line did not appear feasible because of unsatisfactory subsoil conditions along the waterfront.

Another factor which influenced the design in considerable measure was the small margin for allowable pressure losses between the Hudson Ave. boilers and the throttle of the Gold St. turbines. For this reason alone it was imperative to reach the objective in the most direct manner in order to deliver the specified steam quantity at the required pressure with a single line and readily available pipe sizes.

The solution that satisfied these conditions involved the use of overhead support by means of two suspension cables. The pipe run between the two generating stations was divided into three spans by installing two intermediate towers in such locations as to cause the least interference with the traffic problems of the plant.

The cables are 1¼ in. diameter galvanized steel bridge cables consisting of six strands of 19 wires each, with a wire rope center. Their ultimate capacity is approximately 124 tons. The suspenders are 1 in. diameter eye bars with turnbuckles. At their lower end the suspenders engage inverted U-bolts which in turn carry the hanger assembly of the steam and the condensate return line (Fig. 4). This sketch also shows a 1 in. heating line carrying 620 F steam installed under the condensate line as a protection against freezing.

The generous use of linked joints has the purpose of insuring an independent and unrestrained expansion of the two lines. The necessity for this arrangement becomes evident if one considers that the rate of the two expansions is as one to six and that the motions in one span are of opposite direction.

The hanger rods were erected inclined to the vertical; that is, their lower ends were offset one-half the anticipated expansion movement in respect to the point of suspension on the cable. Thus the deviation from the plumb line during hot and cold condition was reduced to a minimum.

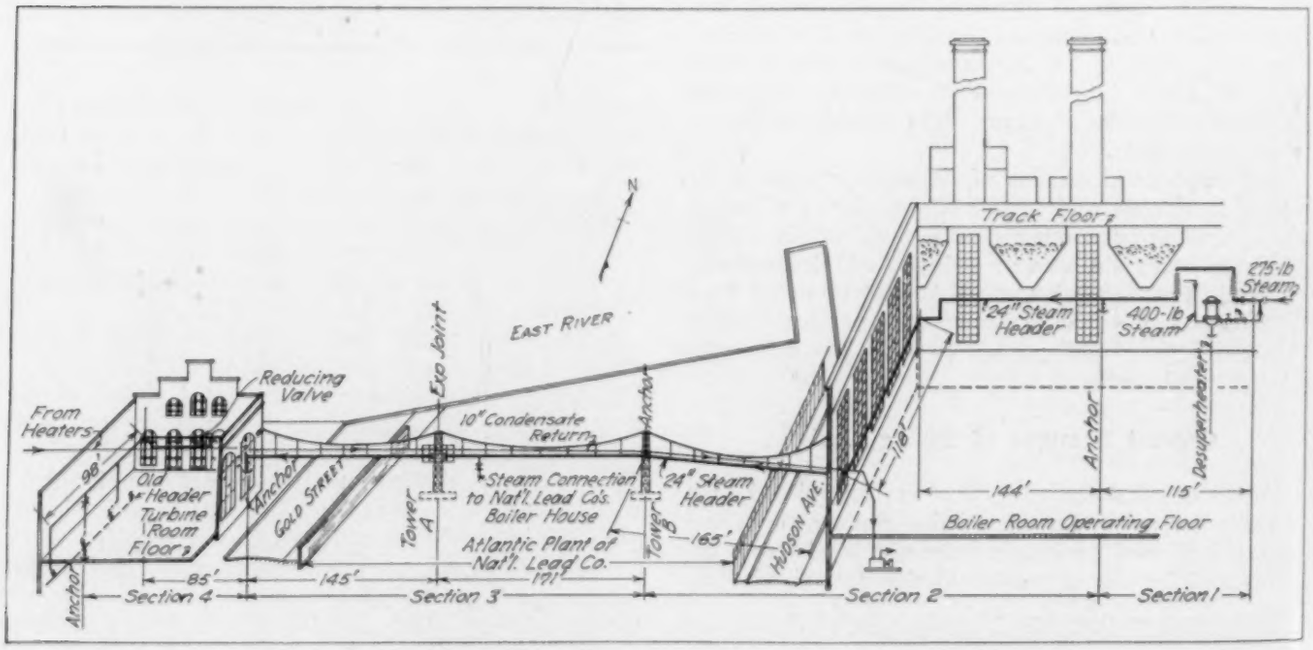
The cables are anchored at each generating station on special steel frames which carry the cable pull to the building foundations.

Load Combinations and Anchorage

The outdoor piping was designed to resist thermal stresses, stresses due to a wind load of 20 lb per sq ft projected on the pipe diameter, and stresses caused by variations of the cable sag at different atmospheric conditions. An analysis of the possible load combinations and resulting pipe deflections brought out the necessity for restraining the ends of each span and of inserting pipe sections of greater wall thickness in the region of negative wind moments. In this manner, a possible deflection of 19 in. due to wind action in the longest span was reduced to 3¼ in.

The following stress values for ¾ in. pipe wall at the ends of the Hudson Ave. span may serve as an illustration of the intensities due to the various actions:

Fig. 2—Diagram of the piping, showing the supporting structures, location of anchors, expansion joint, etc.



from wind, 6600 psi; longitudinal pressure stress, 2050 psi; longitudinal bending stress due to expansion, 5000 psi;—a total of 13,650 psi at the severest working conditions. (psi = lb per sq in.)

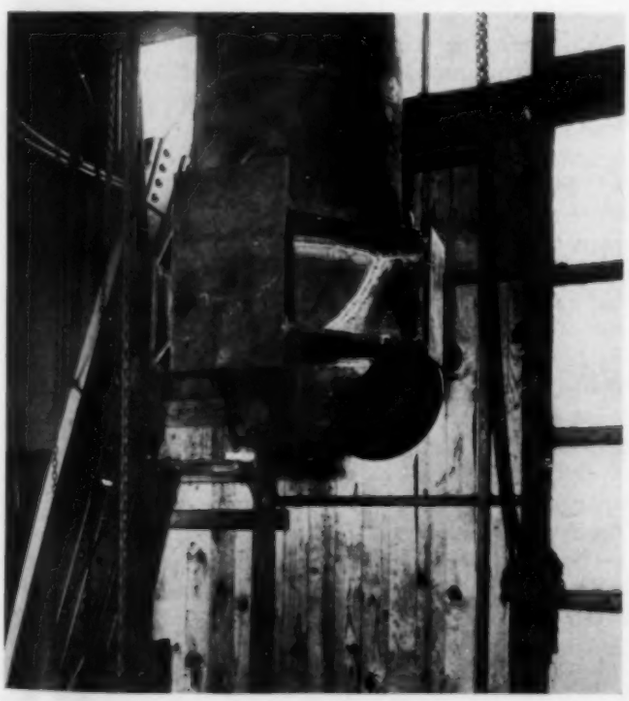
The change in cable sag was estimated at plus or minus 1.5 in. from 65 F at erection to 130 F in the summer and zero temperature in the winter. Additional pipe stresses of 2400 psi due to these variations will occur at fibers located at right angles to those previously mentioned.

The straight run of 316 ft between Tower B and Gold St. Station (Section 3) is protected by an expansion joint at Tower A. The far ends of this section are anchored against a thrust of 115,000 lb. The inner ends are free to expand toward the center of the expansion joint. This section, therefore, carries no longitudinal stress due to expansion but transverse stresses due to steam pressure, which equal (respectively) 6380 psi and 4100 psi in the ½ in. and ¾ in. thick pipe walls. The other stresses caused by wind and change in cable sag are longitudinal and have their maximum in planes which are at right angle to each other.

Expansion Joint

The expansion joint cares for two-thirds of the outdoor piping or an aggregate thermal expansion of about 15 in. The pipe branches which lead into it are anchored at their far ends; *i.e.* at tower B and at the inlet to the Gold St. station. The joint is of the sliding type, that is the pipes entering at both ends with machined surfaces are permitted to expand toward each other by sliding within a steel shell anchored to the tower. Joint tightness is obtained with heat resistant packing material which is forced into stuffing boxes at each entrance. Fittings provide for the lubrication of the packing. The joint is constructed to permit repacking under pressure. To this end, and as a safety measure, two peripheral chambers are grooved into the stationary shell ahead of each stuffing box for the purpose of collecting and discharging steam which may have leaked between the sliding parts.

Fig. 3—Pipe support on steam downtake



Restraining frames were provided at both towers and at the inlet to both stations. At Tower A, the pipe was restrained by means of two outriggers projecting 10 ft from the center of the tower. The outriggers were designed to transfer the moment of restraint to the tower by means of rollers which at the same time guide the pipe into the expansion joint. At Tower B, restraint was provided by special bents at the east and west face of the tower and longitudinal anchorage at the center of the tower. Because of the small amount of expansion from the anchor to these bents, sliding guides were considered adequate for directing the pipe without binding. At the Hudson Ave. outlet, a cantilever structure equipped with roller guides offers the necessary resistance and restraint to wind forces and prevents communication of vibration to the indoor piping.

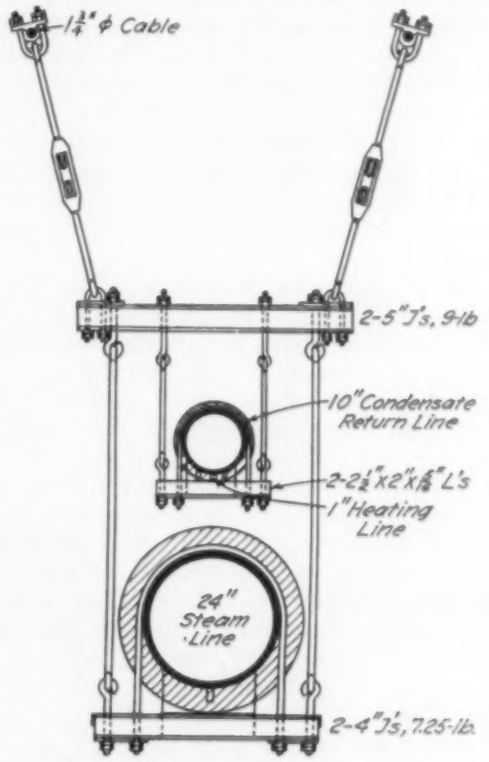


Fig. 4—Hanger for outdoor piping

Erection of the Piping

The outdoor piping was erected from timber bents which carried a continuous working platform for the entire distance between the two stations. The advantage of this erection method became evident as the work progressed and justified the extra cost of the temporary false work. It permitted erecting the greatest part of the piping independently of the tower foundations, the supporting steel and the suspension cables. It facilitated welding, stress relieving and insulating work, and expedited all operations through the ease of communication between the various trade units and the supervisory forces.

[Concluded on p. 239]

Gas Fired Dehumidifier Used for Smoking Fish

THE Mutual Smoked Fish Co., Paterson, N. J., does about $400,000 worth of business a year smoking all manner of fish for New Jersey and metropolitan New York retail outlets. Its seven smokehouses handle 30,000 lb of fish per week, a considerable part of it being mild-cured salmon that is smoked in a new type of smokehouse in which dehumidified air from a gas fired dehumidifying unit does the drying. The only fire in the smokehouse is that required to keep a few oak logs smouldering and giving off the flavoring smoke.

"The biggest danger in this business," comments Mutual's chief smoker, "is the 'burning' or 'overcooking' of fish. If a smokehouse gets too hot the fish get mushy, bad looking, and fragile. When the drying is done with dehumidified air which is never allowed to exceed 90 F in temperature, the likelihood of overcooking is reduced."

The installation was made last July. The smokehouse measures 25 by 25 ft in floor area, 20 ft high, and has a volume of 12,500 cu ft. It is supplied with 1300 cfm of fresh, dried air at 40 to 45 per cent relative humidity and 80 to 90 F dry bulb. The air comes from a silica gel dehumidifying unit fired with city gas.

Operation is continuous and control is automatic. The gas burners of the unit use 275 cfh of 525 Btu per cu ft gas. There is a connected load of 2 kw for operating the blower and other devices which are part of the unit. After the air is dried it is generally considerably warmer (usually 150 F) than the 80 to 90 F required. Therefore, it is passed through a finned cooling coil carrying water at 45 to 60 F.

A. Block, president of the Mutual Smoked Fish Co., racks choice chubbs—smoked in one of his seven gas fired smokehouses

For humidity control, a humidistat is situated inside a duct through which a small sample of air from the smokehouse itself is continually drawn. Should the humidity tend to rise above 45 per cent, the humidistat acts to make the blower pull more air through the silica gel beds and less air direct from the outside. When the smokehouse atmosphere approaches 40 per cent, the humidistat acts in the opposite direction to cause less air to flow through the gel beds and more air to be drawn direct from outside.

A new method of smoking mild-cured salmon, utilizing dehumidified air supplied by a gas fired silica gel unit, is employed by the Mutual Smoked Fish Co. of Paterson, N. J. . . . . Use of dehumidified air instead of charcoal fires is said to have a number of advantages in reducing cost and improving quality of the product. The system is described here by Harry W. Smith, Jr.*

The smokehouse is a brick walled room with an insulated roof. All windows are painted to exclude sun heat. The drying air, controlled as to moisture and temperature, comes in in sheet metal duct work which runs horizontally along one wall about 5 ft from the floor and enters the room through three registers. A damper control regulates the flow. Circulation in the smokehouse is achieved mainly by a propeller type circulator fan about 6 ft above the inlet registers, although desk fans are used at times in various spots throughout the smokehouse to keep the air moving correctly in accordance with the distribution of fish loaded into the chamber. The exhaust opening from the room is at the top, 20 ft up, in the same wall on which the inlet registers are situated.

An interesting feature of the installation is that the dehumidifying unit and cooler are in a sheet metal penthouse on the roof of the plant, and thus take no plant floor space. Such a convenient set-up is possible only because the unit requires little or no attention. It need only be turned on when the fish are loaded in the smokehouse and the flow of dry air is needed, and turned off when the job is done. Otherwise its operation is automatic and this is the reason for a large part of the saving that can be attributed to the method. It eliminates the cost of laying and lighting smokehouse fires, constantly checking smokehouse temperatures and adjusting heat inputs accordingly, and in cleaning up when the cycle is complete. A. Block, president of the company, summarizes the advantages as follows: "Faster drying with consequent increased capacity, greater convenience, better quality of fish, greater uniformity of product, less trouble, and lower cost."

The smokehouse is used four times per week, each time with a load of 300 to 400 sides of salmon (8 to 10 lb per side). Previously it took 10 hr to smoke each load—today, with the dry air system, it takes 6 to 8 hr.

*Industrial Gas Section, American Gas Association.

Performance of Coils for Dehumidifying Air

By **William Goodman***

THE appendices for this series of articles on performance of coils for dehumidifying air are continued this month, and will be concluded in May. In this series, the author has developed new and simple formulas for computing the final dry and wet bulb temperatures of the air leaving a coil. Particularly noteworthy is the fact that the final wet bulb depression of the air has been shown to depend upon the initial wet bulb depression of the air for both direct expansion and counterflow coils. This makes computation of the final dry bulb temperature the work of a moment.—THE EDITOR.

Appendix 8

Derivation of Equation 1 for Counterflow Coils

$$H_w = A_w U_w d_M$$

Equation 7 of Appendix 2 is

$$\frac{dH_w}{h - h_R} = U_w dA_w \quad \ldots\ldots [1]$$

In Appendix 2 because h_R is constant for direct expansion coils, the preceding equation leads directly to an integrable form. For counterflow coils, however, h_R is a variable because t_R, the temperature of the refrigerant, varies throughout the coil. Hence this variation in h_R must be considered before [1] can be integrated. Now

$$dH_w = -G dh \quad \ldots\ldots [2]$$

Also, inasmuch as the heat lost by the air is transferred to the refrigerant,

$$dH_w = -G'c' dt_R \quad \ldots\ldots [3]$$

(The negative sign is used in the preceding equation because the temperature of the water falls as the surface increases from left to right in Fig. 9.)

As is evident from the discussions in Appendices 3 and 4, a represents the change in the enthalpy of saturated air per degree change in temperature. Hence,

$$dh_R = a dt_R \quad \ldots\ldots [4]$$

$$dt_R = \frac{dh_R}{a} \quad \ldots\ldots [5]$$

Substitute [5] into [3],

$$dH_w = -\frac{G'c'}{a} dh_R \quad \ldots\ldots [6]$$

From [2],

$$dh = -\frac{dH_w}{G} \quad \ldots\ldots [7]$$

From [6],

$$dh_R = -\frac{a}{G'c'} dH_w \quad \ldots\ldots [8]$$

Subtracting [8] from [7], and rearranging terms,

$$dH_w = -\frac{dh - dh_R}{\left[\frac{1}{G} - \frac{a}{G'c'}\right]} \quad \ldots\ldots [9]$$

$$= -\frac{d(h - h_R)}{\frac{1}{G}\left[1 - \frac{Ga}{G'c'}\right]} \quad \ldots\ldots [10]$$

Substituting [10] into [1] and rearranging terms,

$$\frac{d(h - h_R)}{h - h_R} = -\frac{U_w}{G}\left[1 - \frac{Ga}{G'c'}\right] dA_w \quad \ldots\ldots [11]$$

Equation 11 can now be integrated, yielding

$$\text{Log}_e \frac{h_1 - h_{R2}}{h_2 - h_{R1}} = \frac{A_w U_w}{G}\left[1 - \frac{Ga}{G'c'}\right] \quad \ldots\ldots [12]$$

Referring to Fig. 9,

$$\text{Log}_e \frac{d_L}{d_S} = \frac{A_w U_w}{G}\left[1 - \frac{Ga}{G'c'}\right] \quad \ldots\ldots [13]$$

Equation 24 of the text is obtained by letting

$$M = \frac{A_w U_w}{G}\left[1 - \frac{Ga}{G'c'}\right]$$

But

$$H_w = G(h_1 - h_2) \quad \ldots\ldots [14]$$

and

$$\frac{1}{G} = \frac{h_1 - h_2}{H_w} \quad \ldots\ldots [15]$$

Also

$$G(h_1 - h_2) = G'c'(t_{R1} - t_{R2}) \quad \ldots\ldots [16]$$

Inasmuch as the enthalpy of saturated air depends only upon its temperature, obviously

$$\frac{1}{a}(h_{R1} - h_{R2}) = t_{R1} - t_{R2} \quad \ldots\ldots [17]$$

Substituting [17] into [16],

$$G(h_1 - h_2) = \frac{G'c'}{a}(h_{R1} - h_{R2}) \quad \ldots\ldots [18]$$

$$\frac{Ga}{G'c'} = \frac{h_{R1} - h_{R2}}{h_1 - h_2} \quad \ldots\ldots [19]$$

$$1 - \frac{Ga}{G'c'} = 1 - \frac{h_{R1} - h_{R2}}{h_1 - h_2} \quad \ldots\ldots [20]$$

Multiplying [20] by [15], clearing, and reassembling,

$$\frac{1}{G}\left[1 - \frac{Ga}{G'c'}\right] = \frac{1}{H_w}\left[(h_1 - h_{R2}) - (h_2 - h_{R1})\right] \quad [21]$$

$$= \frac{1}{H_w}(d_L - d_S) \quad \ldots\ldots [22]$$

Substituting [22] into [13],

$$\text{Log}_e \frac{d_L}{d_S} = \frac{A_w U_w}{H_w}(d_L - d_S) \quad \ldots\ldots [23]$$

$$H_w = A_w U_w \left[\frac{d_L - d_S}{\text{log}_e \frac{d_L}{d_S}}\right] \quad \ldots\ldots [24]$$

Let

$$d_M = \frac{d_L - d_S}{\text{log}_e \frac{d_L}{d_S}} \quad \ldots\ldots [25]$$

Therefore,

$$H_w = A_w U_w d_M \quad \ldots\ldots [26]$$

This is Equation 1 of the text. This same equation can be derived by using Fig. 11 instead of Fig. 9.

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*The Trane Co. Member of Board of Consulting and Contributing Editors.

Part 6. Part 1 was published in November, 1938, pp. 697-701 and 707; Part 2 in December, 1938, pp. 777-781; Part 3 in January, 1939, pp. 13-19; Part 4 in February, 1939, pp. 83-86; and Part 5 in March, pp. 157-160.

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Appendix 9

Derivation of Equation 23

$$H_w = \frac{G\,(h_1 - h_{R1})}{1 + \dfrac{G}{A_w U_w F}}$$

For $\dfrac{Ga}{G'c'} < 1$ (Refer to Fig. 9)

From Equation 1 of the text,

$$d_M = \frac{H_w}{A_w U_w} \quad \ldots\ldots[1]$$

$$\frac{d_M}{d_s} = \frac{H_w}{A_w U_w d_s} \quad \ldots\ldots[2]$$

Let

$$F = \frac{d_M}{d_s} \quad \ldots\ldots[3]$$

$$F = \frac{H_w}{A_w U_w d_s} \quad \ldots\ldots[4]$$

$$d_s = \frac{H_w}{A_w U_w F} \quad \ldots\ldots[5]$$

Referring to Fig. 9,

$$d_s = h_2 - h_{R1} \quad \ldots\ldots[6]$$

$$h_2 - h_{R1} = \frac{H_w}{A_w U_w F} \quad \ldots\ldots[7]$$

For $\dfrac{Ga}{G'c'} > 1$ (Refer to Fig. 11)

$$d_M = \frac{H_w}{A_w U_w} \quad \ldots\ldots[1a]$$

$$\frac{d_M}{d_L} = \frac{H_w}{A_w U_w d_L} \quad \ldots\ldots[2a]$$

Let

$$F = \frac{d_M}{d_L} \quad \ldots\ldots[3a]$$

$$F = \frac{H_w}{A_w U_w d_L} \quad \ldots\ldots[4a]$$

$$d_L = \frac{H_w}{A_w U_w F} \quad \ldots\ldots[5a]$$

Referring to Fig. 11,**

$$d_L = h_2 - h_{R1} \quad \ldots\ldots[6a]$$

$$h_2 - h_{R1} = \frac{H_w}{A_w U_w F} \quad \ldots\ldots[7a]$$

**Note: d_L and d_s were interchanged in error on Fig. 11.

From here on the derivation for both cases is identical. It is only necessary to bear in mind the fact that the value of F is different for the two cases.

$$H_w = G\,(h_1 - h_2) \quad \ldots\ldots[8]$$

$$h_2 = h_1 - \frac{H_w}{G} \quad \ldots\ldots[9]$$

Substitute [9] into [7], and solving for H_w,

$$H_w = \frac{G\,(h_1 - h_{R1})}{1 + \dfrac{G}{A_w U_w F}}$$

This is Equation 23 of the text.

Appendix 10

Derivation of Equation 8 for Counterflow Coils

$$\frac{D_2}{D_1} = e^{-M}$$

Equation 5 of Appendix 5 is

$$\frac{1}{U_w}\,\frac{t - t_R}{h - h_R} = \frac{1}{f_g}\,\frac{dH_s}{dH_w} + \frac{B}{f_R} \quad \ldots\ldots[1]$$

In the case of counterflow coils, h_R and t_R are variables because the temperature of the refrigerant changes as it flows through the coil. This variation must be considered in casting [1] into an integrable form.

$$dH_s = -Gc\,dt \quad \ldots\ldots[2]$$

$$dt = -\frac{dH_s}{Gc} \quad \ldots\ldots[3]$$

$$dH_w = G'c'\,dt_R \quad \ldots\ldots[4]$$

$$dt_R = -\frac{dH_w}{G'c'} \quad \ldots\ldots[5]$$

Subtract [5] from [3],

$$dt - dt_R = -\frac{dH_s}{Gc} + \frac{dH_w}{G'c'} \quad \ldots\ldots[6]$$

$$\frac{d\,(t - t_R)}{dH_w} = -\frac{1}{Gc}\,\frac{dH_s}{dH_w} + \frac{1}{G'c'} \quad \ldots\ldots[7]$$

$$\frac{dH_s}{dH_w} = \frac{Gc}{G'c'} - Gc\,\frac{d\,(t - t_R)}{dH_T} \quad \ldots\ldots[8]$$

Substitute [8] into [1],

$$\frac{1}{U_w}\,\frac{t - t_R}{h - h_R} = \frac{1}{f_g}\,\frac{Gc}{G'c'} - \frac{Gc}{f_g}\,\frac{d\,(t - t_R)}{dH_T} + \frac{B}{f_R} \quad \ldots\ldots[9]$$

But [10] of Appendix 8 is

$$dH_w = -\frac{d\,(h - h_R)}{\dfrac{1}{G}\left[1 - \dfrac{Ga}{G'c'}\right]} \quad \ldots\ldots[10]$$

Substitute [10] into [9] and rearrange terms,

$$\frac{d(t - t_R)}{d(h - h_R)} - \left[\frac{f_g}{cU_w\left[1 - \frac{Ga}{G'c'}\right](h - h_R)}\right](t - t_R) = -\frac{\frac{B}{c}\,\frac{f_g}{f_R} + \frac{G}{G'c'}}{\left[1 - \frac{Ga}{G'c'}\right]} \quad \ldots\ldots[11]$$

This is a linear differential equation of the form $\dfrac{dy}{dx} + Py = Q$.

The solution of [11]* is

$$\frac{a(t_2 - t_{R1}) - (h_2 - h_{R1})}{a(t_1 - t_{R2}) - (h_1 - h_{R2})} = \left[\frac{h_1 - h_{R2}}{h_2 - h_{R1}}\right]^{-\frac{f_g}{cU_w\left[1 - \frac{Ga}{G'c'}\right]}} \quad \ldots\ldots[12]$$

From Equation 12 of Appendix 8,

$$\frac{h_1 - h_{R2}}{h_2 - h_{R1}} = e^{\frac{A_w U_w}{G}\left[1 - \frac{Ga}{G'c'}\right]} \quad \ldots\ldots[13]$$

$$\left[\frac{h_1 - h_{R2}}{h_2 - h_{R1}}\right]^{-\frac{f_g}{cU_w\left[1 - \frac{Ga}{G'c'}\right]}} = e^{-\frac{f_g A_w}{Gc}} = e^{-M} \quad \ldots\ldots[14]$$

(By Equation 11 of the text, $M = f_g A_w / Gc$).

Substituting [14] into [12]

$$\frac{a(t_2 - t_{R1}) - (h_2 - h_{R1})}{a(t_1 - t_{R2}) - (h_1 - h_{R2})} = e^{-M} \quad \ldots\ldots[15]$$

Inasmuch as the enthalpy of an air-vapor mixture is very nearly constant for any given wet bulb temperature,

$$h_2 - h_{R1} = a(t'_2 - t_{R1}) \quad \ldots\ldots[16]$$

$$h_1 - h_{R2} = a(t'_1 - t_{R2}) \quad \ldots\ldots[17]$$

Substituting [16] and [17] into [15], and rearranging,

$$\frac{t_2 - t'_2}{t_1 - t'_1} = e^{-M} \quad \ldots\ldots[18]$$

Substituting [18] and [19] of Appendix 5,

$$\frac{D_2}{D_1} = e^{-M} \quad \ldots\ldots[19]$$

This is Equation 8 of the text.

Appendix 11

Derivation of Equation 25

$$D_B = \frac{(1 + R)D'' + ND_D}{R + N}$$

*In solving Equation 11, the left-hand term of the equation below occurs. By substituting Equation 3 of the text for U_w in the denominator of the left-hand term, it can be reduced to the simple term on the right-hand side.

$$\frac{\dfrac{Bf_g}{cf_R} + \dfrac{G}{G'c'}}{\left[1 - \dfrac{Ga}{G'c'}\right]\left[1 - \dfrac{f_g}{cU_w\left(1 - \dfrac{Ga}{G'c'}\right)}\right]} = \frac{1}{a}$$

a = a numerical coefficient. See Table 1.
A_D = external area of dry surface, sq ft.
A_T = total external area of coil surface, sq ft.
A_W = external area of wetted surface, sq ft.
B = ratio of external to internal surface area of coil.
c = humid specific heat of air (average $c = 0.243$).
c' = specific heat of refrigerant.
d_L = large enthalpy (total heat) difference, Btu.
d_M = mean enthalpy (total heat) difference, Btu.
d_S = small enthalpy (total heat) difference, Btu.
D = wet bulb depression of the air, deg Fahr.
$D'' = t''_1 - t_R$, or $t''_1 - t_{R2}$.
$D_R = t_R - t_R$, or $t_R - t_{R2}$.
$D_D = t_1 - t_{R2}$.
D_L = large temperature difference, deg Fahr.
D_M = mean temperature difference, deg Fahr.
D_S = small temperature difference, deg Fahr.
f_a = coefficient of heat transfer through air film, Btu per hr per sq ft per degree of temperature difference.
f_R = coefficient of heat transfer through refrigerant film, Btu per hr per sq ft per degree of temperature difference.
$F = \frac{d_M}{d_S}$, or $\frac{d_M}{d_L}$. See text.
G = weight of air, lb per hr.
G' = weight of refrigerant, lb per hr.
h = enthalpy (total heat) of humid air, Btu per lb of dry air.
h_R = enthalpy (total heat) of saturated air at a temperature equal to the refrigerant temperature, Btu per lb of dry air.
h_S = enthalpy (total heat) of saturated air at a temperature equal to the surface temperature, Btu per lb of dry air.
H_D = total heat transferred through dry surface of coil, Btu per hr.
H_S = sensible heat lost by the air flowing past the wetted surface, Btu per hr.
H_W = total heat lost by the air flowing past the wetted surface, Btu per hr.
M = a numerical factor. See Equations 7, 11, and 24.
$N = \frac{Gc}{G'c'}$. $R = \frac{Bf_a}{f_R}$.
t = dry bulb temperature, deg Fahr.
t' = wet bulb temperature, deg Fahr.
t'' = dew point temperature, deg Fahr.
t_R = boundary dry bulb temperature, deg Fahr.
t'_R = boundary wet bulb temperature, deg Fahr.
t_R = refrigerant temperature, deg Fahr.
t_{RB} = temperature of the refrigerant at the boundary, deg Fahr.
t_S = surface temperature of coil, deg Fahr.
U_D = over-all coefficient of heat transfer through dry surface of coil, Btu per hr per sq ft per degree of temperature difference.
U_W = over-all coefficient of heat transfer through wetted surface of coil, Btu per hr per sq ft per Btu of enthalpy difference.
V_a = air velocity over face area of coil, fpm.
V_w = water velocity through tubes of coil, fps.
θ = angle in Fig. 4.

The subscript "1" refers to the initial condition of the air or refrigerant entering the coil. The subscript "2" refers to the final condition of the air or refrigerant leaving the coil. All logarithms in formulas are to the base e.

Equation 6 of Appendix 7 is,

$$\frac{t_R - t''_1}{t''_1 - t_R} = \frac{f_R}{Bf_a} \quad \ldots\ldots [1]$$

Inasmuch as the temperature of the refrigerant changes in a counterflow coil, t_R is, in this case, the temperature of the refrigerant at the boundary. Hence [1] may be rewritten

$$\frac{t_R - t''_1}{t''_1 - t_{RB}} = \frac{f_R}{Bf_a} \quad \ldots\ldots [2]$$

Substituting [13] of Appendix 7,

$$t_R = \left[1 + \frac{1}{R}\right] t''_1 - \frac{t_{RB}}{R} \quad \ldots\ldots [3]$$

Inasmuch as this equation contains two unknown variables (t_R and t_{RB}) one of them must be eliminated. t_{RB} can be eliminated by means of Equation 30 of the text. From Equation 30 of the text,

$$t_{RB} = t_{R2} - N(t_1 - t_R) \quad \ldots\ldots [4]$$

where

$$N = \frac{Gc}{G'c'} \quad \ldots\ldots [5]$$

Substituting [4] into [3], and rearranging,

$$\left[\frac{N}{R} + 1\right] t_R = \left[1 + \frac{1}{R}\right] t''_1 - \frac{t_{R2}}{R} + N\frac{t_1}{R} \quad \ldots\ldots [6]$$

$$\left[\frac{N}{R} + 1\right] t_R - \left[\frac{N}{R} + 1\right] t_{R2} =$$

$$\left[1 + \frac{1}{R}\right] t''_1 - \frac{t_{R2}}{R} + N\frac{t_1}{R} - \left[\frac{N}{R} + 1\right] t_{R2} \quad \ldots\ldots [7]$$

$$\left[\frac{N}{R} + 1\right](t_R - t_{R2}) = \left[1 + \frac{1}{R}\right](t''_1 - t_{R2}) + \frac{N}{R}(t_1 - t_{R2}) \quad \ldots\ldots [8]$$

Let

$$D_R = t_R - t_{R2} \quad \ldots\ldots [9]$$
$$D'' = t''_1 - t_{R2} \quad \ldots\ldots [10]$$
$$D_D = t_1 - t_{R2} \quad \ldots\ldots [11]$$

$$\left[\frac{N}{R} + 1\right] D_R = \left[1 + \frac{1}{R}\right] D'' + \frac{N}{R} D_D \quad \ldots\ldots [12]$$

$$(R + N) D_R = (1 + R) D'' + N D_D \quad \ldots\ldots [13]$$

$$D_R = \frac{(1 + R) D'' + N D_D}{R + N} \quad \ldots\ldots [14]$$

This is Equation 25 of the text.

Appendix 12

Condition Curve when f_a Is Infinite

Equation 6 of Appendix 7 is

$$\frac{t_R - t''_1}{t''_1 - t_R} = \frac{f_R}{Bf_a} \quad \ldots\ldots [1]$$

If either f_a or B is infinite, or if $f_R = 0$, Equation 1 reduces to

$$t_R = t''_1 \quad \ldots\ldots [2]$$

In other words, dehumidification of the air cannot take place until the dry bulb temperature of the air has been lowered to the dew point temperature—that is, the air is cooled to saturation.

Appendix 13

Condition Curve when f_R Is Infinite

Equation 3 of Appendix 7 may be written

$$\frac{t_S - t_R}{t - t_S} = \frac{Bf_a}{f_R} \quad \ldots\ldots [1]$$

When f_R is infinite, or if either B or f_a equals zero, Equation 1 reduces to

$$t_S = t_R \quad \ldots\ldots [2]$$

For this condition, the surface temperature is constant throughout the coil, and equal to the refrigerant temperature. Keevil and Lewis† have shown that if the surface temperature is constant, the condition of the air during cooling by a wetted surface can

†Keevil and Lewis, *Industrial and Engineering Chemistry*, Vol. 20, p. 1058, October, 1928.

be represented on the psychrometric chart by a straight line connecting the initial condition of the air with the constant surface temperature—in this case, the refrigerant temperature.

Appendix 14

Condition Curve when A_W Is Infinite

Direct Expansion Coils

When A_W is infinite, Equation 10 of Appendix 2 reduces to

$$h_2 = h_R \quad \ldots\ldots [1]$$

The final wet bulb temperature of the air is equal to the constant refrigerant temperature.

To find the final dry bulb temperature of the air leaving an infinite surface, Equation 17 of Appendix 3 must be used. When A_W is infinite this equation reduces to

$$t_2 = t'_2 \quad \ldots\ldots [2]$$

The final dry bulb temperature of the air is also equal to the constant refrigerant temperature. Hence the air will always be cooled to saturation by an infinite surface. This does not necessarily mean that an infinite surface is required to cool air to saturation, but only that an infinite surface will always cool air to saturation.

Counterflow Coils (Ga/G'c' less than one)

For this condition, when A_W is infinite, Equation 12 of Appendix 8 reduces to

$$h_2 = h_{R1} \quad \ldots\ldots [3]$$

The final wet bulb temperature of the air will be equal to the initial refrigerant temperature.

For this condition, when A_W is infinite, Equation 8 of the text reduces to

$$t_2 = t'_2 \quad \ldots\ldots [4]$$

Therefore, the final dry bulb temperature of the air is also equal to the initial temperature of the refrigerant. Hence the air will always be cooled to saturation by an infinite surface.

Counterflow Coils (Ga/G'c' greater than one)

For this condition, when A_W is infinite, Equation 12 of Appendix 8 reduces to

$$h_1 = h_{R2} \quad \ldots\ldots [5]$$

In other words, in this case the final temperature of the refrigerant will be equal to the initial wet bulb temperature of the air. To find the final wet bulb temperature of the air leaving an infinite surface, Equation 31 of the text must be used.

For this condition, when A_W is infinite, Equation 8 of the text reduces to

$$t_2 = t'_2 \quad \ldots\ldots [6]$$

In this case also, the final dry bulb temperaure of the air is equal to the final wet bulb temperature, and the air is therefore saturated.

[To be concluded]

Air Conditioning Eliminates Afternoon Fatigue

With a complete heating and cooling season now behind them, officials of the Commercial Travelers Mutual Accident Association of America report that the usual afternoon fatigue experienced by employees in their building at Utica, N. Y., has practically been eliminated. "From that one item alone we feel that our year 'round air conditioning system has justified itself," declares Edward Trevvett, assistant secretary-treasurer.

In 1936 Commercial Travelers decided to remodel their office building in Utica and construct a large addition on the adjoining lot. Plans were drawn by Gauge and Ames, architects, and A. R. Acheson, consulting engineer, planned and issued specifications for air conditioning which was installed by Hudson Air Conditioning Corp.

Because of the characteristics of the building, and the changing solar and wind effects on various parts of it, the system was zoned to maintain even temperatures throughout winter and summer. The building was divided into five separate year 'round conditioning zones and one straight winter conditioning zone, with direct radiation used at such points as entries and lavatories.

A vault presented a difficult problem because no openings were permitted into it through which ducts could be run. The vault consists of two floors, the first one in the basement. In order to circulate conditioned air through it, air was discharged in a closed hallway into which the basement vault door opens. A brick duct was constructed at the rear corner of the vault, with two openings in the basement room, and an opening at the ceiling of the second floor room to accommodate a 12 in. fan. With this arrangement, and the vault doors left open during working hours, it is possible to circulate conditioned air through the basement vault, up the duct into the room above, and out into air conditioned space. Because of the small load involved, a small conditioner was used, with fans mounted on the inside and driven by a 1 hp motor. K. G. PATRICK.*

*General Electric Co.

Properties of Air-Vapor Mixtures Interpolated to Tenths of a Degree

By William Goodman*

THE tables on the next three pages are concluded from the ones published in January, February and March; with the January table appeared a brief explanation of them.

The values in Columns 3 and 4 are extremely useful for quickly finding the *actual* enthalpy (total heat) of an air-vapor mixture if its dry bulb and dew point temperatures are known. A complete discussion was given in the January, 1938, HPAC[1] of the method of using the actual enthalpy of the air for accurate computations as against the approximate methods commonly used. Although the error is generally not large, there are times when accurate computations using the actual enthalpy should be used. In such cases the equation below, which was developed in the earlier article, offers the quickest means of finding the actual enthalpy if the dry bulb and dew point temperatures are known.

$$h = s'' t + 1061w$$

where h = actual enthalpy of the mixture, Btu per lb of dry air;
t = dry bulb temperature of the air.

Values of s'', the humid specific heat of air corresponding to any given *dew point* temperature, are tabulated in Column 3. Values of $1061w$ corresponding to the *dew point* temperatures are tabulated in Column 4.

Example: Find the actual enthalpy of a quantity of air if its dry bulb temperature is 90.5 deg and its dew point temperature is 65.3 deg.

Solution: Referring to the table, for a dew point temperature of 65.3 deg, $s'' = 0.2460$, and $1061w = 14.19$.

$$h = s'' t + 1061w$$
$$= 0.246 \times 90.5 + 14.19$$
$$= 36.45 \text{ Btu per lb of dry air, actual enthalpy}$$

*The Trane Co. Member of Board of Consulting and Contributing Editors.

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[1]"New Tables of the Psychrometric Properties of Air-Vapor Mixtures." HEATING, PIPING AND AIR CONDITIONING, January, 1938, pp. 1-7.

Absolute Humidity and Enthalpy of Mixtures of Air and Saturated Water Vapor

Interpolated to Tenths of a Degree.
Barometric Pressure 14.696 Lb per Sq In.
All Tabulated Values are Quantities per Pound of Dry Air in the Mixture.

Temp. Deg Fahr t	Absolute Humidity Pounds w	Humid Specific Heat s'	$1061w$	Enthalpy of Air-Vapor Mixture Btu h	Temp. Deg Fahr t
1	2	3	4	5	6
65.0	0.01323	0.2460	14.04	30.02	65.0
.1	.01328		14.09	30.10	.1
.2	.01332		14.14	30.17	.2
.3	.01337		14.19	30.25	.3
.4	.01342		14.24	30.32	.4
65.5	0.01347	0.2461	14.29	30.40	65.5
.6	.01351		14.34	30.48	.6
.7	.01356		14.39	30.55	.7
.8	.01361		14.44	30.63	.8
.9	.01365		14.49	30.70	.9
66.0	0.01370	0.2462	14.54	30.78	66.0
.1	.01375		14.59	30.86	.1
.2	.01380		14.64	30.94	.2
.3	.01385		14.70	31.02	.3
.4	.01390	0.2463	14.75	31.10	.4
66.5	0.01395	0.2463	14.80	31.18	66.5
.6	.01399		14.85	31.25	.6
.7	.01404		14.90	31.33	.7
.8	.01400		14.96	31.41	.8
.9	.01414	0.2464	15.01	31.49	.9
67.0	0.01419	0.2464	15.06	31.57	67.0
.1	.01424		15.11	31.65	.1
.2	.01429		15.17	31.73	.2
.3	.01434	0.2465	15.22	31.81	.3
.4	.01439		15.28	31.89	.4
67.5	0.01445	0.2465	15.33	31.97	67.5
.6	.01450		15.38	32.05	.6
.7	.01455	0.2466	15.44	32.13	.7
.8	.01460		15.49	32.21	.8
.9	.01465		15.55	32.29	.9
68.0	0.01470	0.2466	15.60	32.37	68.0
.1	.01475		15.66	32.45	.1
.2	.01481	0.2467	15.71	32.53	.2
.3	.01486		15.77	32.62	.3
.4	.01491		15.82	32.70	.4
68.5	0.01497	0.2467	15.88	32.78	68.5
.6	.01502	0.2468	15.94	32.86	.6
.7	.01507		15.99	32.94	.7
.8	.01512		16.05	33.03	.8
.9	.01518		16.10	33.11	.9
69.0	0.01523	0.2469	16.16	33.19	69.0
.1	.01528		16.22	33.27	.1
.2	.01534		16.27	33.36	.2
.3	.01539		16.33	33.44	.3
.4	.01545	0.2470	16.39	33.53	.4
69.5	0.01550	0.2470	16.45	33.61	69.5
.6	.01555		16.50	33.69	.6
.7	.01561		16.56	33.78	.7
.8	.01566	0.2471	16.62	33.86	.8
.9	.01572		16.67	33.95	.9
70.0	0.01577	0.2471	16.73	34.03	70.0
.1	.01583		16.79	34.12	.1
.2	.01588	0.2472	16.85	34.20	.2
.3	.01594		16.91	34.29	.3
.4	.01599		16.97	34.37	.4
70.5	0.01605	0.2472	17.03	34.46	70.5
.6	.01611	0.2473	17.09	34.55	.6
.7	.01616		17.15	34.63	.7
.8	.01622		17.21	34.72	.8
.9	.01627		17.27	34.80	.9
71.0	0.01633	0.2474	17.33	34.89	71.0
.1	.01639		17.39	34.98	.1
.2	.01645		17.45	35.07	.2
.3	.01650		17.51	35.15	.3
.4	.01656	0.2475	17.57	35.24	.4
71.5	0.01662	0.2475	17.64	35.33	71.5
.6	.01668		17.70	35.42	.6
.7	.01674		17.76	35.51	.7
.8	.01679	0.2476	17.82	35.59	.8
.9	.01685		17.88	35.68	.9
72.0	0.01691	0.2476	17.94	35.77	72.0
.1	.01697		18.00	35.86	.1
.2	.01703	0.2477	18.07	35.95	.2
.3	.01709		18.13	36.04	.3
.4	.01715		18.20	36.13	.4
72.5	0.01721	0.2477	18.26	36.22	72.5
.6	.01727	0.2478	18.32	36.31	.6
.7	.01733		18.39	36.40	.7
.8	.01739		18.45	36.49	.8
.9	.01745	0.2479	18.52	36.58	.9
73.0	0.01751	0.2479	18.58	36.67	73.0
.1	.01757		18.65	36.76	.1
.2	.01763		18.71	36.85	.2
.3	.01770	0.2480	18.78	36.95	.3
.4	.01776		18.84	37.04	.4

Temp. Deg Fahr t	Absolute Humidity Pounds w	Humid Specific Heat s'	$1061w$	Enthalpy of Air-Vapor Mixture Btu h	Temp. Deg Fahr t
1	2	3	4	5	6
73.5	0.01782	0.2480	18.91	37.13	73.5
.6	.01788		18.98	37.22	.6
.7	.01794	0.2481	19.04	37.31	.7
.8	.01801		19.11	37.41	.8
.9	.01807		19.17	37.50	.9
74.0	0.01813	0.2482	19.24	37.59	74.0
.1	.01819		19.31	37.68	.1
.2	.01826		19.37	37.78	.2
.3	.01832		19.44	37.87	.3
.4	.01838	0.2483	19.50	37.97	.4
74.5	0.01845	0.2483	19.57	38.06	74.5
.6	.01851		19.64	38.15	.6
.7	.01857	0.2484	19.70	38.25	.7
.8	.01863		19.77	38.34	.8
.9	.01870		19.83	38.44	.9
75.0	0.01876	0.2484	19.90	38.53	75.0
.1	.01883	0.2485	19.97	38.63	.1
.2	.01889		20.04	38.72	.2
.3	.01896		20.11	38.82	.3
.4	.01902	0.2486	20.18	38.92	.4
75.5	0.01909	0.2486	20.26	39.02	75.5
.6	.01916		20.33	39.11	.6
.7	.01922	0.2487	20.40	39.21	.7
.8	.01929		20.47	39.31	.8
.9	.01935		20.54	39.40	.9
76.0	0.01942	0.2487	20.61	39.50	76.0
.1	.01949	0.2488	20.68	39.60	.1
.2	.01956		20.75	39.70	.2
.3	.01962		20.83	39.80	.3
.4	.01969	0.2489	20.90	39.90	.4
76.5	0.01976	0.2489	20.97	40.00	76.5
.6	.01983		21.04	40.09	.6
.7	.01990	0.2490	21.11	40.19	.7
.8	.01996		21.19	40.29	.8
.9	.02003		21.26	40.39	.9
77.0	0.02010	0.2490	21.33	40.49	77.0
.1	.02017	0.2491	21.40	40.59	.1
.2	.02024		21.48	40.69	.2
.3	.02031		21.55	40.79	.3
.4	.02038	0.2492	21.62	40.89	.4
77.5	0.02045	0.2492	21.70	41.00	77.5
.6	.02051		21.77	41.10	.6
.7	.02058	0.2493	21.84	41.20	.7
.8	.02065		21.91	41.30	.8
.9	.02072		21.99	41.40	.9
78.0	0.02079	0.2494	22.06	41.50	78.0
.1	.02086		22.14	41.60	.1
.2	.02093		22.21	41.71	.2
.3	.02101	0.2495	22.29	41.81	.3
.4	.02108		22.36	41.92	.4
78.5	0.02115	0.2495	22.44	42.02	78.5
.6	.02122	0.2496	22.52	42.12	.6
.7	.02129		22.59	42.23	.7
.8	.02137		22.67	42.33	.8
.9	.02144		22.74	42.44	.9
79.0	0.02151	0.2497	22.82	42.54	79.0
.1	.02158		22.90	42.65	.1
.2	.02166		22.98	42.75	.2
.3	.02173	0.2498	23.06	42.86	.3
.4	.02181		23.14	42.96	.4
79.5	0.02188	0.2498	23.22	43.07	79.5
.6	.02195	0.2499	23.29	43.18	.6
.7	.02203		23.37	43.28	.7
.8	.02210		23.45	43.39	.8
.9	.02218	0.2500	23.53	43.49	.9
80.0	0.02225	0.2500	23.61	43.60	80.0
.1	.02233		23.69	43.71	.1
.2	.02240	0.2501	23.77	43.82	.2
.3	.02248		23.85	43.93	.3
.4	.02256	0.2502	23.93	44.04	.4
80.5	0.02264	0.2502	24.02	44.15	80.5
.6	.02271		24.10	44.26	.6
.7	.02279	0.2503	24.18	44.37	.7
.8	.02287		24.26	44.48	.8
.9	.02294		24.34	44.59	.9
81.0	0.02302	0.2504	24.42	44.70	81.0
.1	.02310		24.50	44.81	.1
.2	.02318		24.59	44.92	.2
.3	.02326	0.2505	24.67	45.04	.3
.4	.02334		24.76	45.15	.4
81.5	0.02342	0.2505	24.84	45.26	81.5
.6	.02349	0.2506	24.92	45.37	.6
.7	.02357		25.01	45.48	.7
.8	.02365		25.09	45.60	.8
.9	.02373	0.2507	25.18	45.71	.9

[See preceding page]

Absolute Humidity and Enthalpy of Mixtures of Air and Saturated Water Vapor

Interpolated to Tenths of a Degree.
Barometric Pressure 14.696 Lb per Sq In.
All Tabulated Values are Quantities per Pound of Dry Air in the Mixture.

Temp. Deg Fahr t	Absolute Humidity Pounds w	Humid Specific Heat s''	$1061w$	Enthalpy of Air-Vapor Mixture Btu h	Temp. Deg Fahr t
1	2	3	4	5	6
82.0	0.02381	0.2507	25.26	45.82	82.0
.1	.02389	0.2508	25.35	45.93	.1
.2	.02397		25.43	46.05	.2
.3	.02406		25.52	46.16	.3
.4	.02414	0.2509	25.61	46.28	.4
82.5	0.02422	0.2509	25.70	46.39	82.5
.6	.02430		25.78	46.50	.6
.7	.02438	0.2510	25.87	46.62	.7
.8	.02447		25.96	46.73	.8
.9	.02455		26.04	46.85	.9
83.0	0.02463	0.2511	26.13	46.96	83.0
.1	.02471		26.22	47.08	.1
.2	.02480	0.2512	26.31	47.19	.2
.3	.02488		26.39	47.31	.3
.4	.02496		26.48	47.43	.4
83.5	0.02505	0.2513	26.57	47.55	83.5
.6	.02513		26.66	47.66	.6
.7	.02521		26.75	47.78	.7
.8	.02529	0.2514	26.83	47.90	.8
.9	.02538		26.92	48.01	.9
84.0	0.02546	0.2515	27.01	48.13	84.0
.1	.02555		27.10	48.25	.1
.2	.02563		27.20	48.37	.2
.3	.02572	0.2516	27.29	48.49	.3
.4	.02581		27.38	48.61	.4
84.5	0.02590	0.2517	27.48	48.73	84.5
.6	.02598		27.57	48.85	.6
.7	.02607		27.66	48.97	.7
.8	.02616	0.2518	27.75	49.09	.8
.9	.02624		27.85	49.21	.9
85.0	0.02633	0.2519	27.94	49.33	85.0
.1	.02642		28.04	49.45	.1
.2	.02651		28.13	49.58	.2
.3	.02660	0.2520	28.23	49.70	.3
.4	.02669		28.32	49.83	.4
85.5	0.02678	0.2521	28.42	49.95	85.5
.6	.02687		28.51	50.07	.6
.7	.02696		28.61	50.20	.7
.8	.02705	0.2522	28.70	50.32	.8
.9	.02714		28.80	50.45	.9
86.0	0.02723	0.2523	28.89	50.57	86.0
.1	.02732		28.99	50.70	.1
.2	.02741		29.09	50.82	.2
.3	.02751	0.2524	29.18	50.95	.3
.4	.02760		29.28	51.07	.4
86.5	0.02769	0.2525	29.38	51.20	86.5
.6	.02778		29.48	51.33	.6
.7	.02787		29.58	51.45	.7
.8	.02797	0.2526	29.67	51.58	.8
.9	.02806		29.77	51.70	.9
87.0	0.02815	0.2527	29.87	51.83	87.0
.1	.02824		29.97	51.96	.1
.2	.02834	0.2528	30.07	52.09	.2
.3	.02843		30.17	52.22	.3
.4	.02853		30.27	52.35	.4
87.5	0.02862	0.2529	30.37	52.48	87.5
.6	.02871		30.46	52.60	.6
.7	.02881	0.2530	30.56	52.73	.7
.8	.02890		30.66	52.86	.8
.9	.02900		30.76	52.99	.9
88.0	0.02909	0.2531	30.86	53.12	88.0
.1	.02919		30.96	53.25	.1
.2	.02929	0.2532	31.07	53.39	.2
.3	.02938		31.17	53.52	.3
.4	.02948	0.2533	31.28	53.66	.4
88.5	0.02958	0.2533	31.38	53.79	88.5
.6	.02968	0.2534	31.48	53.92	.6
.7	.02978		31.59	54.06	.7
.8	.02987		31.69	54.19	.8
.9	.02997	0.2535	31.80	54.33	.9
89.0	0.03007	0.2535	31.90	54.46	89.0
.1	.03017	0.2536	32.01	54.60	.1
.2	.03027		32.12	54.73	.2
.3	.03037	0.2537	32.22	54.87	.3
.4	.03047		32.33	55.00	.4
89.5	0.03058	0.2538	32.44	55.14	89.5
.6	.03068		32.55	55.28	.6
.7	.03078	0.2539	32.66	55.41	.7
.8	.03088		32.76	55.55	.8
.9	.03098		32.87	55.68	.9
90.0	0.03108	0.2540	32.98	55.82	90.0
.1	.03118		33.09	55.96	.1
.2	.03129	0.2541	33.20	56.10	.2
.3	.03139		33.31	56.24	.3
.4	.03150	0.2542	33.42	56.38	.4
90.5	0.03160	0.2542	33.53	56.52	90.5
.6	.03170	0.2543	33.64	56.66	.6
.7	.03181		33.75	56.80	.7
.8	.03191	0.2544	33.86	56.94	.8
.9	.03202		33.97	57.08	.9
91.0	0.03212	0.2545	34.08	57.22	91.0
.1	.03223		34.19	57.36	.1
.2	.03233		34.31	57.51	.2
.3	.03244	0.2546	34.42	57.65	.3
.4	.03255		34.54	57.79	.4
91.5	0.03266	0.2547	34.65	57.94	91.5
.6	.03276		34.76	58.08	.6
.7	.03287	0.2548	34.88	58.22	.7
.8	.03298		34.99	58.36	.8
.9	.03308	0.2549	35.11	58.51	.9
92.0	0.03319	0.2549	35.22	58.65	92.0
.1	.03330	0.2550	35.34	58.80	.1
.2	.03341		35.45	58.95	.2
.3	.03352	0.2551	35.57	59.09	.3
.4	.03363		35.69	59.24	.4
92.5	0.03375	0.2552	35.81	59.39	92.5
.6	.03386		35.92	59.54	.6
.7	.03397	0.2553	36.04	59.69	.7
.8	.03408		36.16	59.83	.8
.9	.03419	0.2554	36.27	59.98	.9
93.0	0.03430	0.2554	36.39	60.13	93.0
.1	.03441	0.2555	36.51	60.28	.1
.2	.03453		36.63	60.43	.2
.3	.03464	0.2556	36.75	60.58	.3
.4	.03476		36.87	60.73	.4
93.5	0.03487	0.2557	37.00	60.89	93.5
.6	.03498		37.12	61.04	.6
.7	.03510	0.2558	37.24	61.19	.7
.8	.03521		37.36	61.34	.8
.9	.03533	0.2559	37.48	61.49	.9
94.0	0.03544	0.2560	37.60	61.64	94.0
.1	.03556		37.72	61.80	.1
.2	.03567	0.2561	37.85	61.95	.2
.3	.03579		37.97	62.11	.3
.4	.03591	0.2562	38.10	62.26	.4
94.5	0.03603	0.2562	38.22	62.42	94.5
.6	.03614	0.2563	38.34	62.57	.6
.7	.03626		38.47	62.73	.7
.8	.03638	0.2564	38.59	62.88	.8
.9	.03649		38.72	63.04	.9
95.0	0.03661	0.2565	38.84	63.19	95.0
.1	.03673		38.97	63.35	.1
.2	.03685	0.2566	39.10	63.51	.2
.3	.03697		39.23	63.67	.3
.4	.03709	0.2567	39.36	63.83	.4
95.5	0.03722	0.2568	39.49	63.99	95.5
.6	.03734		39.61	64.14	.6
.7	.03746	0.2569	39.74	64.30	.7
.8	.03758		39.87	64.46	.8
.9	.03770	0.2570	40.00	64.62	.9
96.0	0.03782	0.2570	40.13	64.78	96.0
.1	.03794	0.2571	40.26	64.94	.1
.2	.03807		40.39	65.10	.2
.3	.03819	0.2572	40.52	65.27	.3
.4	.03832		40.65	65.43	.4
96.5	0.03844	0.2573	40.79	65.59	96.5
.6	.03856	0.2574	40.92	65.75	.6
.7	.03869		41.05	65.91	.7
.8	.03881	0.2575	41.18	66.08	.8
.9	.03894		41.31	66.24	.9
97.0	0.03906	0.2576	41.44	66.40	97.0
.1	.03919		41.58	66.57	.1
.2	.03932	0.2577	41.71	66.74	.2
.3	.03945	0.2578	41.85	66.91	.3
.4	.03958		41.99	67.08	.4
97.5	0.03971	0.2579	42.13	67.25	97.5
.6	.03983		42.26	67.41	.6
.7	.03996	0.2580	42.40	67.58	.7
.8	.04009		42.54	67.75	.8
.9	.04022	0.2581	42.67	67.92	.9
98.0	0.04035	0.2582	42.81	68.09	98.0
.1	.04048		42.95	68.26	.1
.2	.04062	0.2583	43.09	68.43	.2
.3	.04075		43.23	68.60	.3
.4	.04088	0.2584	43.37	68.77	.4
98.5	0.04102	0.2585	43.52	68.95	98.5
.6	.04115		43.66	69.12	.6
.7	.04128	0.2586	43.80	69.29	.7
.8	.04141		43.94	69.46	.8
.9	.04155	0.2587	44.08	69.63	.9

[See second preceding page]

Absolute Humidity and Enthalpy of Mixtures of Air and Saturated Water Vapor

Interpolated to Tenths of a Degree.
Barometric Pressure 14.696 Lb per Sq In.
All Tabulated Values are Quantities per Pound of Dry Air in the Mixture.

Temp. Deg Fahr t	Absolute Humidity Pounds w	Humid Specific Heat s'	$1061w$	Enthalpy of Air-Vapor Mixture Btu h	Temp. Deg Fahr t
1	2	3	4	5	6
99.0	0.04168	0.2588	44.22	69.80	99.0
.1	.04182		44.37	69.98	.1
.2	.04195	0.2589	44.51	70.15	.2
.3	.04209		44.66	70.33	.3
.4	.04222	0.2590	44.80	70.51	.4
99.5	0.04236	0.2591	44.95	70.69	99.5
.6	.04250		45.09	70.86	.6
.7	.04263	0.2592	45.24	71.04	.7
.8	.04277		45.38	71.22	.8
.9	.04290	0.2593	45.53	71.39	.9
100.0	0.04304	0.2594	45.67	71.57	100.0

[See third preceding page]

Wells—

[Concluded from p. 228]

often used in connection with quality, which is not accurate as some artesian waters are not fit to use for any purpose.

As a rule waters from shallow or deep wells above the rock formations are softer than waters from deep rock wells drilled into the rock formations. Upper waters stand more chance of being contaminated than deep waters, while deep waters are generally more highly mineralized than upper waters.

The temperature of ground water at the surface is usually within 2 deg higher or lower than 56 F and increases roughly 1 deg for every 150 ft in depth; this is approximately correct for most of the country. It can be more definitely stated as follows: the ground water temperature at the surface is about the same as the average daily temperature of the preceding year.

Reclaiming Failed Wells

Engineers should take with a pinch of salt statements made concerning restoration or reclamation of failed wells. Various methods are used to do this, such as back blowing with air, gas, steam, water or dry ice. Where the screens are of material that will stand acid treatment, this seems to be the most promising and good results have been had in many cases. Some pressure methods such as surging and excessive pumping have been helpful, but all well work of this kind depends largely on the skill and experience of the well man and results vary accordingly.

Scientific well building is an art requiring a good working knowledge of geology, hydraulics, and chemistry coupled with a high degree of mechanical skill. Not all well drillers have these qualifications but a large number of them do—or at least know where to get advice on their technical problems. The whole trade interested in ground water development is constantly seeking new methods of construction and better materials which will make for better wells.

Steam Tie Line—

[Concluded from p. 231]

The steam pipe was lifted in 30 ft sections, set to exact elevation, and welded in place. At the completion of all pipe joints, the hangers were installed and adjusted to calculated dimensions, following a predetermined sequence which intended and actually accomplished a gradual transfer of the pipe load to the cables without producing unbalanced cable forces at the tops of the towers.

Insulation

In selecting the insulation for the outdoor piping, consideration was given to structural strength of the material, adaptability to weatherproofing, ability to recover insulating efficiency after accidental wetting, over-all economy in cost of material, application and maintenance. A sponge felted insulation, constructed from felts of asbestos fibers and spongy cellular material, furnished integral with a weatherproofed jacket of three ply asphalt impregnated felt was selected. Erection economies were possible because it could be obtained in sectional form with integral weatherproofing in the 24 in. size. After weatherproofing the laps on the jacket with asphalt cement, the insulation was fastened in place with galvanized iron bands, drawn tight and clinched in place.

The same material was chosen for the condensate return line for the further reason that the sponge felt, being fabricated of rolled laminations, could be built to the exact circumference required to encase both the condensate line and the 1 in. steam heating line adjacent to it.

Knowledge of All Conditions Essential

W. D. Gerber, engineer of the Illinois State Water Survey, pointed out in his paper on air conditioning water supply and disposal at the conference at the University of Illinois last month that cities definitely have authority to prohibit disposal of waste water into sanitary sewers. He cited the case of a theater which drove a well to supply water for air conditioning, installed the necessary equipment, and then discovered that there was no place to get rid of the water as the city engineer would not permit putting it into the sanitary sewer. There was, however, a storm sewer two blocks away and the theater solved the problem by installing its own pipe line to the storm sewer. Mr. Gerber used the case to stress the importance of determining beforehand not only the method of water supply, but the method of water disposal.

Return Well Clogged

In another case, a return well was dug for the purpose of disposing of waste water. This example, too, showed the importance of a thorough knowledge of all factors involved as after the return well was put in use, it was found that iron in the water oxidized to form a jelly-like substance which clogged up the well in 30 days.

Water Spray Ring Reduces Fly Ash

By Edward E. James*

A METHOD of combating fly ash which has been in use for about three years at the Cincinnati General Hospital has reduced complaints approximately 95 per cent. The device is a water spray ring installed in the chimney as shown in the diagram.

The chimney is 250 ft high with an 8 ft top and a 12 ft bottom inside diameter. When this stack was erected the boilers were equipped with V-type furnaces with dutch ovens and operated with natural draft. Later, the furnaces were remodeled and underfeed stokers with forced draft were installed, and the high stack draft caused complaints from the neighborhood about fly ash.

After some experimenting the mechanical department constructed what is called a water spray ring. Three lengths of 3/4 in. brass pipe were bent into a circle, using the 12 ft flywheel of an engine as a guide. Three-sixty-fourth in. holes 1 in. apart were then drilled a little off center on the inside of the ring so the spray would be downward and to the center of the ring.

When the drilling of the holes was completed, the ring was erected temporarily in the boiler room about 15 ft above the floor. Tests were made, and when we had the spray meeting in the center of the ring we marked each joint; after dismantling the ring and assembling it in the chimney care was taken to have all joints as they were when we had it connected up temporarily.

The three pieces of pipe were placed inside the base of the chimney and connected with unions; a ¾x¾x1½ in. brass tee was placed in the ring for the water supply connection.

The erection was made with some difficulty as the flue gas currents made work almost impossible. This difficulty was overcome by placing electric fans on the chimney floor and blowing the flue gases upward; the men also covered their mouths and noses with cloths. The ring was raised 5 ft and attached with U-bolts to 1½ in. angle iron 5 ft long, this process continuing until the ring was 4 ft above the top of the breeching. Only three angle iron supports were used, these being placed at equal distance around the ring. Each time we raised the ring 10 ft we connected a 10 ft length of 1½ in. brass pipe to the ring.

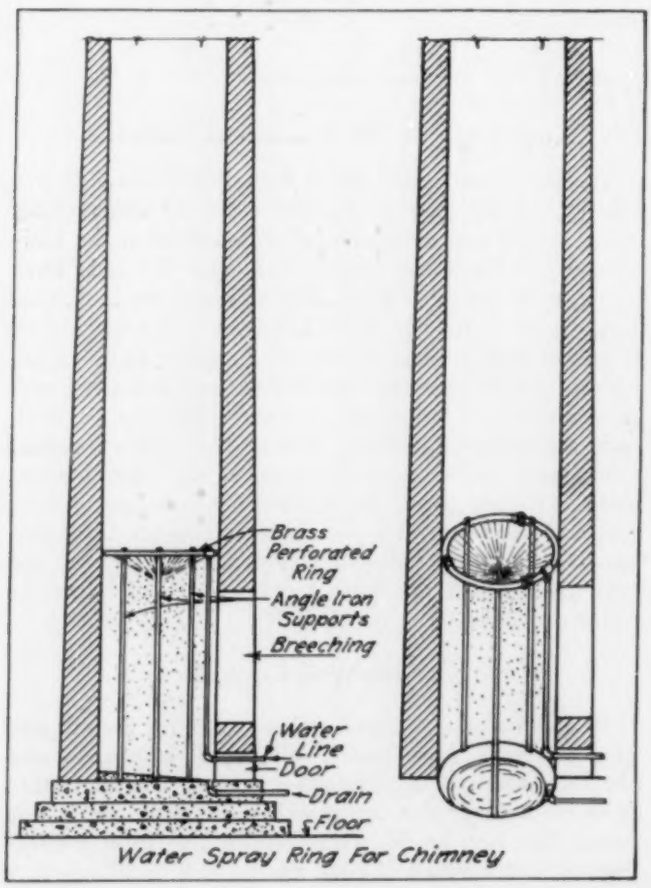
When the installation was completed and tried out, several inches of water was left standing in the bottom of the chimney. We then poured a concave concrete floor sloping to the door where we had installed a 4 in. drain line running to a settling tank in the boiler room. This tank has an overflow to the sewer, allowing the water to flow away and the fly ash to remain in the tank, which is cleaned out each time the spray is used. The condition in our plant made the installation ideal as the chimney floor is 7 ft above the boiler room floor.

To prevent the small holes in the ring from scaling and possibly stopping up, we use water from our zeolite softener. The plant operates 24 hours a day, three 8 hr watches, and each watch blows the boiler tubes once. The water is turned on the spray just before the fireman starts to blow the boiler tubes and is turned off a short time after he has finished blowing the tubes.

About four bushels of fly ash is collected each time the tubes are blown. The spray ring has been in operation about three years and our complaints have been reduced about 95 per cent, as the chimney emits practically no solids when the men are blowing the tubes.

*Chief Engineer, Cincinnati General Hospital.

When fly ash from the chimney of the Cincinnati General Hospital caused complaints from the neighbors, the mechanical department devised the water spray ring described here to combat the nuisance. . . . . It has been in use for about three years, and has practically eliminated all of these complaints



Water Spray Ring For Chimney

Oxygen is produced commercially by the liquid air process which involves temperatures as low as 310 deg below zero. Acetylene is derived from calcium carbide, itself formed in the electric furnace. Burning together, these gases produce a flame which can unite or sever metals with equal ease by oxy-acetylene welding or cutting.

Heating · Piping · Air Conditioning

AMERICAN SOCIETY of HEATING and VENTILATING ENGINEERS

Journal Section

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After a careful investigation and report by the Publication Committee on the wide range of subjects which are offered to the Society for discussion and publication, the Council has concluded that a revision of publication policy is advisable to meet the expanding interests of the membership, so that a wider selection of subjects may be presented, and a more concise presentation of reports can be made.

The 12 fundamental characteristics established for acceptable types of papers remain unchanged. The branches of the art of heating, ventilating and air conditioning, as revealed by research, have been enlarged and new rules governing the preparation and presentation of data have been prepared. Members are invited to present any original work dealing with the art of heating, ventilating and air conditioning, new technique, or unique treatment of mathematical problems, designs, operations, etc., for the enlightenment of their fellow members.

Papers for original publication in the monthly JOURNAL, and presentation at meetings, will be acceptable when the text and illustrations do not exceed the equivalent of 5000 words. Where it is desirable to present detailed tables, charts or other data for the scientist or student, or where fundamental research investigations make it essential to have detailed readings for historical reference, these may be presented as an appendix for publication in the TRANSACTIONS.

During the institution of these changes in practice, it will be most helpful to have the full cooperation of the authors of papers.

J. F. McIntire

PRESIDENT.

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Heat Transfer in Storage Water Heaters

By D. W. Nelson* (*MEMBER*) and A. A. Rosenberg,** Madison, Wis.

This paper is the result of research sponsored by the American Society of Heating and Ventilating Engineers in cooperation with the University of Wisconsin.

THE purpose of these tests was to determine capacities and heat transfer coefficients in the common type of horizontal storage water heater. This type of heater usually consists of a horizontal storage tank with an immersed bundle of copper tubes placed horizontally near the bottom. Steam is supplied to the tubes, usually through a motorized valve controlled by an immersion thermostat which is inserted in the side of the tank. In these particular tests, attention was given to the influence of water pressure on the heating capacity.

Description of Apparatus

A photograph of the apparatus used in the test work is shown in Fig. 1, and by reference to the flow diagram in Fig. 2 a comprehensive conception of the setup of the apparatus can be obtained. The storage heater used in the tests was of the horizontal cylindrical type of heavy welded construction, 30 in. in diameter and 60 in. long, having a storage capacity of 185 gal. The heating element consisted of three parallel vertical rows of *U* tubes bent to the proper radius, and rolled into a forged steel tube plate, with a center support provided for the elements. These elements consisted of 16-gage copper tubing having an outside diameter of 1¼ in.; the outside area of the tubes gave a heating surface of 10 sq ft, on the water side. For purposes of inspection and cleaning, the tank was equipped with a manhole cover at the rear and a drain at the bottom.

The heater had two 2-in. tappings: one at the bottom for the cold water inlet, and another on the top of the tank but located at the opposite end for the heated water outlet. Valves were placed on both the inlet and outlet water piping to regulate the supply of water and also its pressure in the tank. A thermometer well was located in the supply piping in order to obtain the temperature of the inlet water. Two additional thermometer wells were placed on the outlet pipe, one directly over the outlet tapping in the heater and the second about a foot downstream. These served as a check on each other in reading the temperature of the outlet water.

The pressure in the tank was indicated by means of a 0 to 100 lb per square inch test gage. The water utilized for the tests was taken from the customary University of Wisconsin water supply which comes from Lake Mendota. This water is used for supplying hot water and for various needs other than for drinking, and has a hardness of about 10 grains per gallon. On

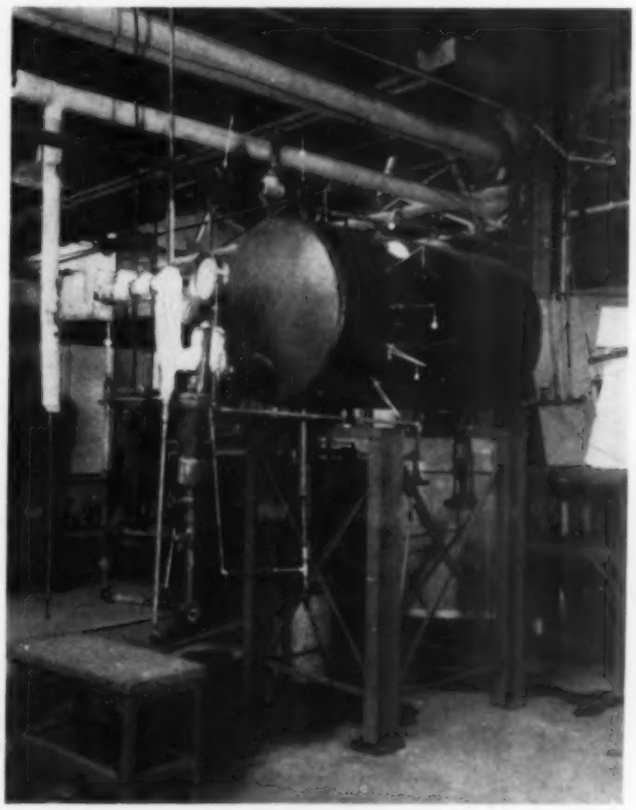

Fig. 1—Photograph of test apparatus

one side of the tank were located six thermometer wells, as shown in Fig. 2, these serving to indicate when the temperatures in the tank were constant during a test. A safety valve was installed in the outlet piping in order to prevent bursting of the tank, due to the expansion of water, should heat be added while the valves were closed. The heated water was conducted through suitable piping to two large tanks of 150-gal capacity each, where the water was alternately weighed on two platform scales.

The steam available for the tests came through a 150-lb per square inch pressure line direct from the University Heating Station, and was brought from the underground piping tunnel first through a 2½-in. line about 90 ft long and then through a 2-in. branch about 40 ft long. The fact that this high pressure steam was generated largely from condensate from the University's heating system means that it probably contained very little air. All of the steam piping up to the inlet to the heater was covered with 2-in. pipe covering. On account of the small pipe sizes there was an appreciable

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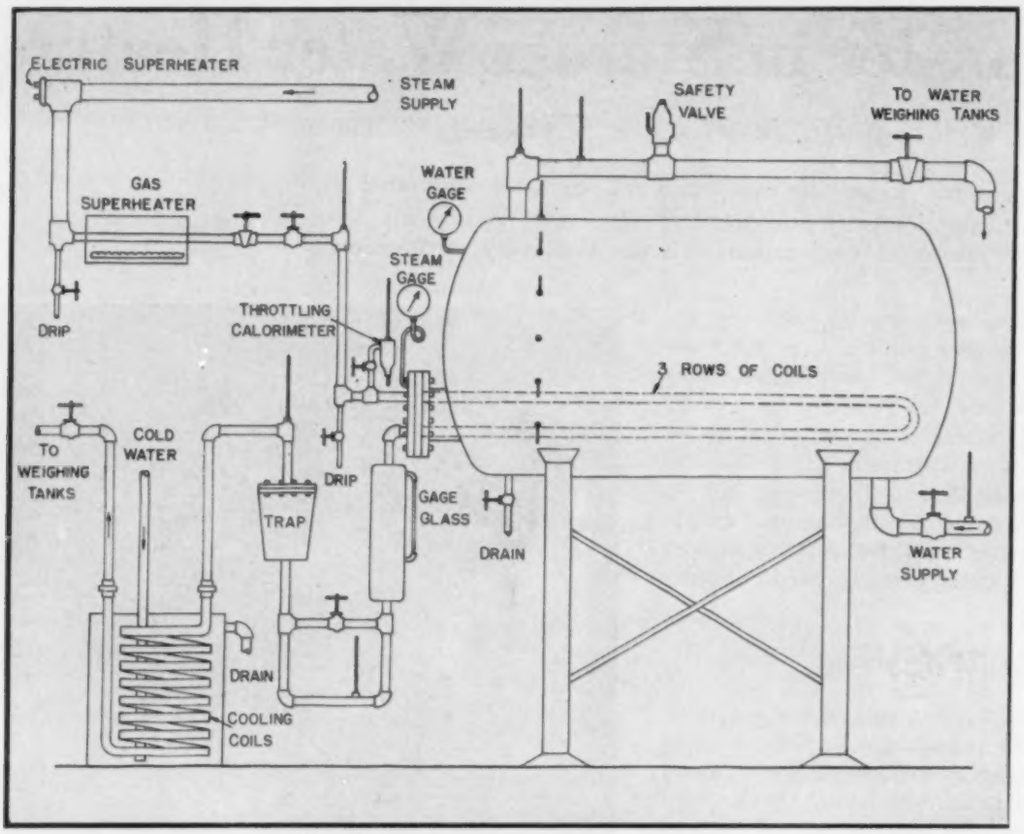

Fig. 2—Diagrammatic sketch of test apparatus

drop in steam pressure, the maximum pressure available during heater tests being about 100 lb per square inch gage.

It was very desirable that the steam entering the tubes in the heater be as nearly dry as possible, or even slightly superheated. Drips were placed at two locations in the steam line before entering the heater. This was done to eliminate the possibility of any condensate being carried with the steam to the heating tubes. In order to dry and slightly superheat the steam, an ordinary gas burner was first used to heat the steam piping a short distance from the heater, as shown in Fig. 2. However, when this proved inadequate, after a number of tests, an improvised electric superheater was installed inside the 2-in. steam supply line. This consisted of nichrome wire supplied with electricity from a 220 AC source.

The flow of steam to the heater was regulated by two hand valves; the first one in the line was a gate valve, and the second was a globe valve. The latter valve served to accomplish the necessary throttling to adjust the steam to the desired pressure. This pressure was measured by means of a 0 to 200 lb per square inch steam gage; also, as a check, and to determine the superheat, if any, the steam temperature was obtained by means of a mercury-in-glass thermometer. The quality or superheat of the steam was determined just before it entered the heater. At the calorimeter, shown in Fig. 2, the temperature was read by a mercury-in-glass thermometer, and the pressure by a mercury manometer.

A receiver equipped with a gage glass was placed directly on the discharge connection from the heater. The outlet level of the condensate was then kept in sight in the gage glass during the tests, with a manually operated globe valve performing this service. In a few of the tests with higher steam pressures, it was found that the trap used was inadequate to handle the flow of condensate. Therefore, in these tests, the trap was removed and a straight section of 1-in. piping was installed; the manually operated valve alone took care of the condensate level in these instances. The condensate from the trap was cooled by passing it through a copper coil which was submerged in running cold water. This precaution was taken to prevent flashing of condensate into steam when discharged at atmospheric pressure. The condensate was alternately weighed in two 50-gal tanks placed on platform scales.

Method of Testing

The duration of a test under any one set of conditions was usually one hour. Before a test was started, the two drips in the steam line were opened, permitting any condensed steam to be eliminated from the system before reaching the heating elements. After connecting the electric superheater to its source of electricity and lighting the gas burners, the steam gate valve was fully opened and the globe valve was partially opened to obtain the desired steam pressure for a particular test. The throttling valve to the calorimeter was opened to give a reading on the manometer of about 1.5 in. of mercury. With the aid of the globe valve located just before the outlet of the condensate, the level of the condensate was maintained at a predetermined point in the gage glass throughout the test.

After the heater had become warm, the water inlet valve was opened, and the supply of water was regulated to obtain the desired outlet temperature. Changing the water supply would also change the steam flow and pressure; so the two valves had to be operated simultaneously. Constancy of readings over a period of 15 min at each of the six surface tank thermometers and of the outlet water temperature served as the criterion for the attainment of equilibrium. A water outlet temperature in the range of 140 to 160 F was chosen for all the tests. The best results were obtained if conditions were

fully adjusted before starting the run, for it was found that adjustments during the test resulted in fluctuations of steam and water pressures, temperatures, and weights. The two pressure gages and the four platform scales were properly calibrated and *stem corrections* were applied to the thermometer readings.

After the desired equilibrium conditions had been obtained, the run was started, and time was recorded by means of a stop watch. Two observers usually conducted the tests, one weighing the water and the other the condensate. Sometimes a third observer assisted in obtaining the various temperature and pressure readings and in regulating valves. At intervals of 10 min throughout the hour of test, the following data were recorded: temperature of water inlet, temperature of water outlet at two points, temperature of steam at two points, temperature and pressure of the throttling calorimeter, pressures of both steam and water, temperatures of condensate before and after the trap, and the weights of both water and condensate. In some of the tests, the quantity of water was of such magnitude as to necessitate the determination of water weights every 5 min. The barometer for each run was also recorded.

Three major series of tests were made: water in the tank under atmospheric pressure (0 lb per square inch gage), water at main pressure (67 to 77 lb per square inch gage), and water at intermediate pressure (between main and atmospheric pressures). In these three series, the steam pressure was varied from 10 to 100 lb per square inch gage, as shown in the summary of results in Tables 1, 2 and 3. In Figs. 3 to 7 inclusive which show the results graphically, circles designate runs

Table 1—Atmospheric Water Pressure Runs

1. Test Number	1*	2*	3*	4*	5	6*	7*	8	9*	10	11*	12*	13	14	15*	16	17	18	19*
2. Steam Pressure, pounds per square inch gage	10	10	15	20	20.1	30	30	39.6	40	40	50	50	60	60	70	79.9	80	80	90
3. Quality Per Cent or Superheat deg F	25F	36F	32F	99.8%	11F	17F	19F	2F	5F	98.0%	5F	5F	98.7%	99.0%	99.8%	99.0%	97.0%	98.8%	99.3%
4. Saturation Steam Temperature	238.5	238.5	248.5	257.8	258.1	273.4	273.5	285.7	286.2	286.3	297.3	297.2	306.9	306.9	315.7	323.6	323.5	323.6	330.9
5. Condensate Temperature, deg F	228.2	226.4	221.7	247.0	252.8	261.1	231.2	278.2	274.9	274.8	288.9	259.3	294.8	297.5	308.1	316.4	272.1	307.5	324.4
6. Condensate Weight, pounds per hour	578	563	746	511	719	1097	1073	963	1165	823	1474	1339	1276	1110	1777	1374	1633	1581	1959
7. Total Heat Supplied, 1000 Btu per hour	565	554	738	485	684	1041	1053	895	1031	751	1360	1275	1155	1005	1605	1220	1498	1418	1738
8. Water Temperature Outlet, deg F	147.9	160.4	158.8	131.7	138.7	147.2	142.7	146.7	147.2	151.0	148.4	148.2	144.7	150.5	148.7	153.0	147.5	160.7	147.7
9. Water Temperature Inlet, deg F	42.0	42.0	42.0	37.5	41.9	42.0	40.0	41.9	43.0	45.0	43.0	47.0	43.0	41.9	41.0	48.1	38.8	36.8	42.0
10. Water Heated, 100 lb per hour	53.1	46.7	64.0	51.3	71.1	99.1	102.8	85.1	100.9	69.9	129.4	125.5	113.7	90.6	149.9	117.9	138.5	114.9	164.7
11. Total Heat Absorbed, 1000 Btu per hour	562	554	747	483	688	1041	1053	890	1050	741	1363	1270	1154	984	1610	1235	1505	1420	1739
12. Radiation and Unaccounted For, 1000 Btu per hour	3	0	—9	2	—4	0	0	5	—19	10	—3	5	1	21	—5	—15	—7	—2	—1
13. Heating Capacity, gallons per hour per square foot per 100 deg rise	67.5	66.5	89.6	57.9	82.5	125.0	126.5	106.5	124.0	88.8	163.5	152.5	138.5	118.0	193.0	148.0	180.5	170.0	208.0
14. Logarithmic Mean Temperature Difference	137	128	140	169	163	175	177	187	186	184	197	195	210	206	216	219	226	219	232
15. Overall Heat Transfer Coefficient, Btu per hour per square foot per deg F	410	433	535	286	422	596	595	476	565	404	693	651	550	478	745	565	668	649	750
16. Water Pressure, pounds per square inch gage	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0

*Cleaned tubes.

Table 2—Intermediate Water Pressure Runs

1. Test Number	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18
2. Steam Pressure, pounds per square inch gage	10	20	20	20	40	40	40	40	40	40	40	60	60	80	80	80	80.1	80.3
3. Quality, Per Cent or Superheat, deg F	99.7%	3F	99.6%	99.4%	98.8%	99.2%	11F	12F	3F	10F	8F	98.5%	98.9%	98.7%	98.5%	98.6%	98.9%	99.0%
4. Saturation Steam Temperature, deg F	238.9	258.1	258.1	258.2	286.5	286.2	286.2	286.2	286.2	286.3	286.3	307.1	306.9	323.6	323.7	323.7	323.7	323.7	323.8
5. Condensate Temperature, deg F	228.1	246.5	248.4	247.9	275.9	273.5	278.9	277.9	277.7	280.5	278.7	295.0	296.9	314.5	241.0	311.5	313.0	313.5	
6. Condensate Weight, pounds per hour	311	350	348	344	454	427	1017	946	808	808	678	583	537	953	592	534	595	533	
7. Total Heat Supplied, 1000 Btu per hour	299	333	329	325	416	395	950	883	752	752	630	526	486	846	570	475	531	475	
8. Water Temperature Outlet, deg F	130.9	137.1	148.7	156.3	161.3	169.4	157.8	147.6	138.1	159.3	146.1	160.1	140.0	148.0	158.3	152.5	133.9	144.7	
9. Water Temperature Inlet, deg F	39.0	39.0	38.0	37.0	38.0	38.0	42.0	42.0	42.0	41.0	42.0	38.0	38.0	41.9	60.9	43.4	42.2	42.5	
10. Water Heated, 100 lb per hour	32.9	32.8	28.4	26.1	32.8	29.3	81.9	83.2	78.2	61.9	59.4	42.9	47.5	81.8	50.6	42.9	60.3	46.6	
11. Total Heat Absorbed, 1000 Btu per hour	302	322	314	312	405	386	946	878	751	733	620	523	485	869	494	468	553	475	
12. Radiation and Unaccounted For, 1000 Btu per hour	—3	11	15	13	11	9	4	5	1	19	10	3	1	—23	—24	7	—22	0	
13. Heating Capacity, gallons per hour per square foot per 100 deg rise	36.2	38.5	37.6	37.4	48.5	46.2	113.5	105.2	90.2	87.9	74.4	62.7	58.2	104.0	59.2	56.1	66.2	57.0	
14. Logarithmic Mean Temperature Difference	150	165	159	154	180	174	180	186	192	180	188	202	214	225	211	222	234	228	
15. Overall Heat Transfer Coefficient, Btu per hour per square foot per deg F	202	195	198	202	225	222	526	472	391	407	330	258	227	386	234	211	236	208	
16. Water Pressure, pounds per square inch gage	25	25	25	45	25	46	5	9	12	14.3	20	25	45	20.6	55	60.3	39.6	61.1	

Table 3—Main Water Pressure Runs

1. Test Number	1	2	3	4	5	6	7	8	9	10	11	12	13	14
2. Steam Pressure, pounds per square inch gage	10.1	20	20	20.4	40	40	60	60	60.4	80	80	80.1	80.4	101.8
3. Quality, Per Cent or Superheat, deg F	98.9%	99.7%	98.6%	98.7%	98.2%	97.5%	98.7%	96.9%	98.1%	98.7%	97.3%	98.1%	98.4%	98.6%
4. Saturation Steam Temperature, deg F	238.8	258.1	258.4	259.0	286.3	286.1	307.1	307.1	307.2	323.5	323.7	323.7	323.7	338.8
5. Condensate Temperature, deg F	230.9	250.6	247.5	251.9	262.3	274.0	290.9	295.3	295.0	311.5	311.3	296.9	311.5	325.8
6. Condensate Weight, pounds per hour	319	317	358	385	470	396	541	477	580	605	604	613	622	649
7. Total Heat Supplied, 1000 Btu per hour	303	299	335	359	435	360	492	424	521	540	531	552	553	570
8. Water Temperature Outlet, deg F	139.8	134.4	121.7	143.1	137.6	151.2	157.2	153.7	147.3	144.5	171.2	150.7	140.9	141.1
9. Water Temperature Inlet, deg F	40.1	43.8	38.0	39.8	38.9	38.0	61.3	40.0	38.9	42.9	40.0	30.9	40.1	40.9
10. Water Heated, 100 lb per hour	29.3	32.3	39.8	31.5	39.9	30.4	50.3	36.6	47.9	50.9	39.3	46.1	51.9	54.4
11. Total Heat Absorbed, 1000 Btu per hour	292	292	333	325	394	344	482	416	519	517	515	515	523	545
12. Radiation and Unaccounted For, 1000 Btu per hour	11	7	2	34	41	16	10	8	2	23	16	37	30	25
13. Heating Capacity, gallons per hour per square foot per 100 deg rise	35.0	35.0	39.9	39.0	47.3	41.3	57.8	49.9	62.3	62.0	61.8	61.8	62.8	65.2
14. Logarithmic Mean Temperature Difference	143	165	175	162	194	186	195	206	209	225	212	224	230	245
15. Overall Heat Transfer Coefficient, Btu per hour per square foot per deg F	20.4	177	190	201	203	185	247	202	248	230	243	230	228	222
16 Water Pressure, pounds per square inch gage	74.3	70	71.3	77.3	75.6	72.5	75	66.9	71	73.8	70	71.5	75.4	74.3

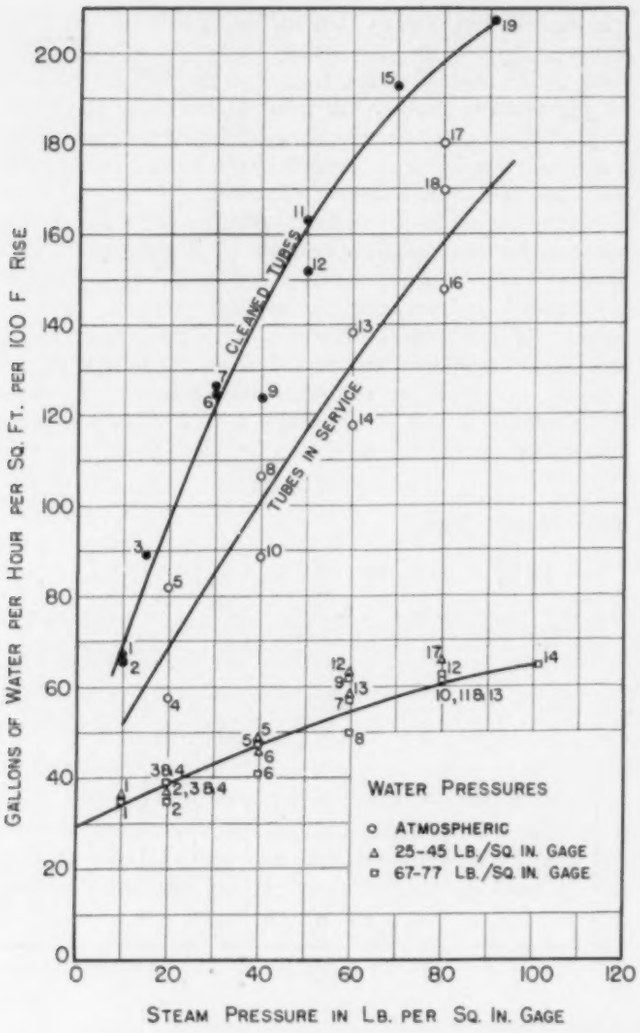
GALLONS OF WATER PER HOUR PER SQ. FT. PER 100 F RISE
STEAM PRESSURE IN LB. PER SQ. IN. GAGE
CLEANED TUBES
TUBES IN SERVICE
WATER PRESSURES
ATMOSPHERIC
25-45 LB./SQ. IN. GAGE
67-77 LB./SQ. IN. GAGE

Fig. 3—Curves showing relationship between water heating capacity and steam pressure

under atmospheric water pressure; squares, runs under main pressure; and triangles, runs under intermediate water pressure; filled-in-circles represent those runs that were made after the tubes had been partially cleaned. This cleaning was done by removing the manhole cover, opening the drain, and then directing a stream of high pressure water on the tubes and on the inside of the tank. The substance removed from the circular bottom of the tank was a rust colored sludge about ½ in. in thickness; a thin coating of soft material was easily removed from the tube surfaces.

Interpretation of Test Results

The principal items recorded for the tests are shown in Tables 1, 2 and 3, and are shown graphically in Figs. 3 to 7 inclusive. In Fig. 3 the capacity in gallons of water heated per hour per square foot per 100 F rise is plotted against steam pressures in pounds per square inch gage. The two upper curves represent the runs made under atmospheric water pressure, whereas the lower curve is for water under higher pressures. Since the points representing intermediate water pressures of 25 to 45 lb per square inch and higher pressures of 67 to 77 lb per square inch gage fall on the same curve, it is evident that the capacity does not vary materially with the water pressure through this range.

Regardless of the amount of water pressure, as the steam pressure increases, the capacity increases; but the increase is at a much greater rate in the case of the atmospheric water pressure runs. At the higher water pressures the capacity increases from 38 to 63 gal per hour, as the steam pressure increases from 20 to 90 lb gage, or an increase in capacity of 66 per cent. Through this same range of steam pressures for the atmospheric runs, the capacity increases from 69 to 170 gal per hour, or an increase of 146 per cent. At 20 lb steam pressure the capacity at atmospheric water pressure is 1.8 times as much as at the higher water pressures; and for the 90 lb steam condition, the capacity is 2.7 times as much.

All of these results were obtained from tubes operating under a condition that might well be encountered in service. With clean tubes, it would be expected that the capacity would increase somewhat, as shown by the uppermost curve on the sheet. After cleaning the tubes, the capacity at 20 lb steam pressure increased 40 per cent, while the capacity at 90 lb steam pressure increased only 21 per cent. Hence, it would appear that the cleaning of tubes would proportionately have a greater effect on increasing the capacity at the lower than at the higher steam pressures.

The overall heat transfer coefficient changes slightly with steam pressure, as shown in Fig. 4, when the water in the tank is maintained at the higher water pressures of 67 to 77 lb per square inch gage, with steam ranging in pressure from 20 to 90 lb the coefficient increases from 190 to 230 Btu per hour per square foot per degree Fahrenheit, or an increase of 21 per cent. Under atmospheric water pressure, for tubes in service, this increase is from 330 to 620 Btu, or an increase of 88 per cent. At 20 lb steam pressure, the lower water pressure gives a heat transfer 1.74 times as much as for the main water pressure; and for the higher steam pressure, the ratio is 2.7 to 1. It will be noted that these

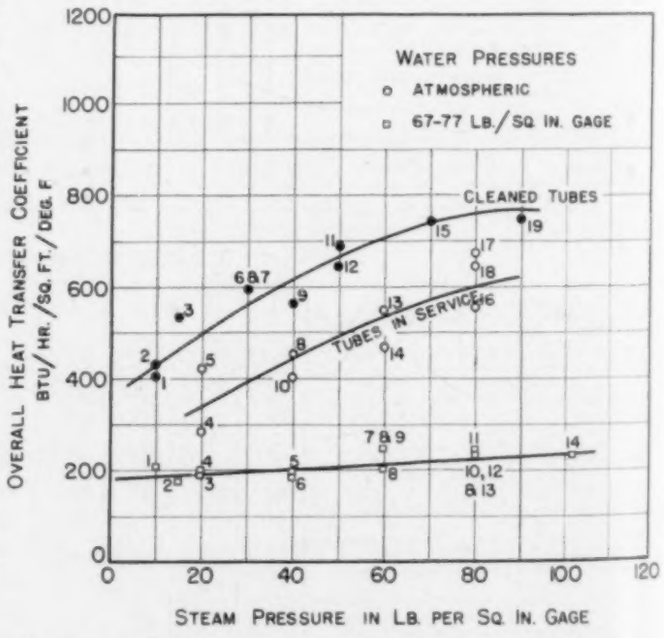

Fig. 4—Overall heat transfer variation with steam pressure

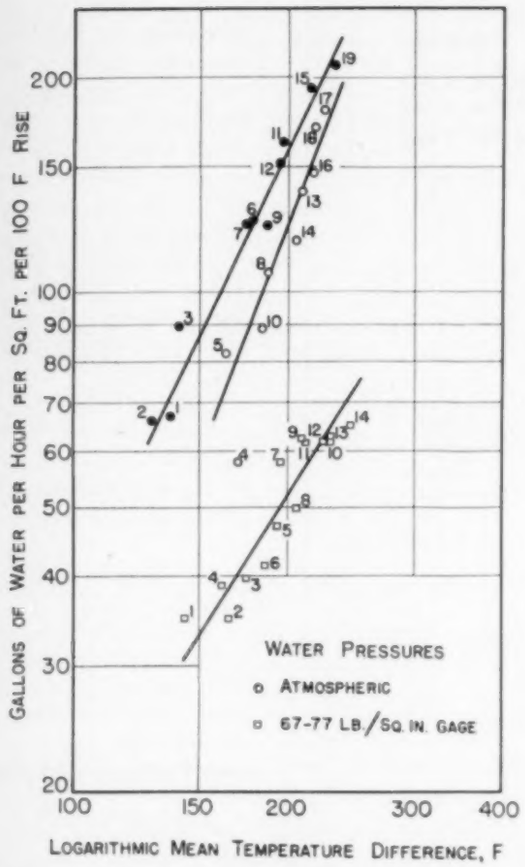
GALLONS OF WATER PER HOUR PER SQ. FT. PER 100 F RISE
LOGARITHMIC MEAN TEMPERATURE DIFFERENCE, F
WATER PRESSURES
ATMOSPHERIC
67-77 LB./SQ. IN. GAGE

Fig. 5—Variation of capacity with logarithmic mean temperature difference plotted on logarithmic coordinates

ratios for the overall heat transfer coefficients are the same as those for capacities mentioned under discussion of Fig. 3, which naturally follows from their interdependency. For cleaned tubes and atmospheric water pressure the coefficient varies from 500 to 770 Btu per hour per square foot per degree Fahrenheit, as the steam pressure is increased from 20 to 90 lb per square inch gage.

The variation of the capacity with the logarithmic mean temperature difference under the two different water pressures is drawn to logarithmic scale in Fig. 5. This figure shows that the water pressure in addition to the mean temperature difference has a large influence in determining the capacity. Under the higher water pressures the capacity varies from 33 to 64 gal of water per hour per square foot per 100 F rise as the temperature difference increases from 150 to 230 F; whereas under atmospheric pressure the capacity increases from 60 to 178 gal through the same range. In the former instance, the increase is about 94 per cent; in the latter, about 197 per cent. At the lower mean temperature difference the capacity is 1.8 times as great for the lower water pressure, while it is 2.8 times as great at the upper temperature difference. Cleaned tubes give a still greater capacity.

Fig. 6 is quite similar to Fig. 5 except that the overall heat transfer coefficient is plotted against the logarithmic mean temperature difference. The large difference that results from the lower water pressure may be readily noted, for the coefficient increases at a much faster rate for the atmospheric pressure runs. With the lower water pressure the coefficient increases from about 310 to 670 Btu per hour per square foot per degree Fahrenheit through the mean temperature difference range of 150 to 230 F, whereas this increase is only from about 190 to 240 Btu with water under main pressure.

The heat transfer coefficient is plotted against water pressure in Fig. 7. Curves are drawn for steam pressures of 10, 40 and 80 lb per square inch gage. The symbols with vertical lines through them represent the runs with 80 lb steam, while those with horizontal lines represent the lowest steam pressure. This figure indicates that for all steam pressures the coefficient and consequently the capacity are not affected as the water pressure is decreased from 75 to about 30 lb per square inch gage. However, as the water pressure decreases from 30 lb to atmospheric pressure, the coefficient increases rapidly. For the lowest of the three steam pressures, the coefficient increases 133 per cent and for the intermediate and highest steam pressures the coefficient increases 154 per cent.

The results clearly show that the capacity of a water heater of this type depends to a considerable extent upon the water pressure in the vessel, the lower pressure giving the greater capacity and heat transfer. Therefore, it was thought desirable that a qualitative explanation be found for this phenomenon; that perhaps a greater turbulence at the lower water pressure was causing the increased heat transfer. One method attempted to prove this phenomenon was to insert a light propellor in the tank to indicate the speed of circulation. The propellor

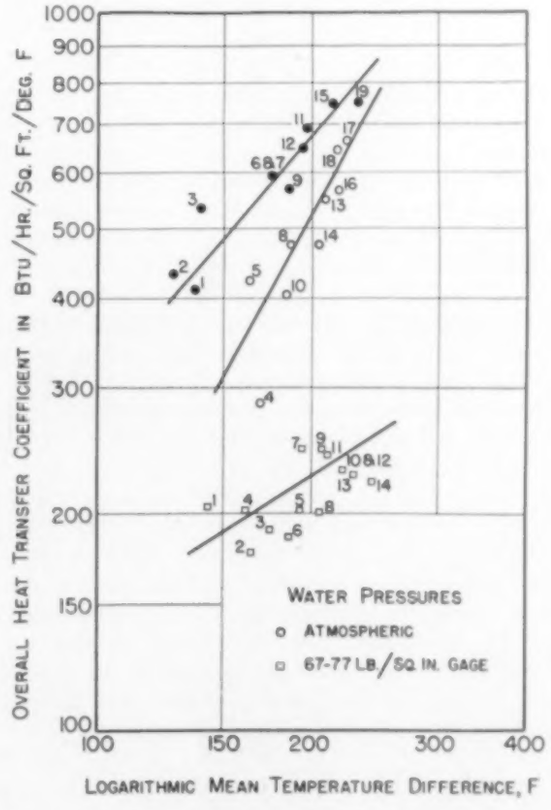
OVERALL HEAT TRANSFER COEFFICIENT IN BTU/HR./SQ. FT./DEG. F
LOGARITHMIC MEAN TEMPERATURE DIFFERENCE, F
WATER PRESSURES
ATMOSPHERIC
67-77 LB./SQ. IN. GAGE

Fig. 6—Curves showing overall heat transfer coefficient against logarithmic mean temperature difference

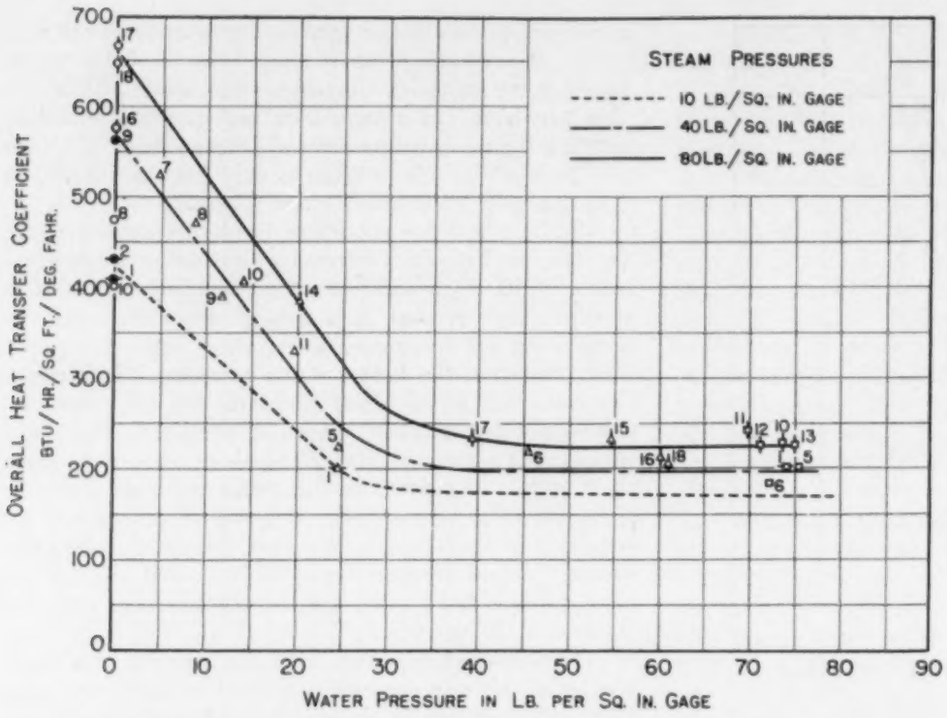

Fig. 7—Graphical relation between heat transfer coefficient and water pressure

was located in the tank directly over the tubes, and was connected to an indicator outside of the tank by means of a ⅛-in. diameter shaft. However, the motion in the tank was not great enough to overcome the friction of the system. Another proposal considered the insertion of a window in the side of the tank, but this was thought to be unsatisfactory for actually observing the action inside the tank near the tube surface.

A third idea, which proved satisfactory, was the duplication on a small scale of conditions inside the heater. The apparatus consisted of a ⅜-in. diameter copper tube placed inside a 1-in. diameter glass tube, the length of these tubes being about 20 in. Steam from the supply line to the heater was allowed to flow through the copper tube, which had a 0 to 50 lb pressure steam gage on the inlet end and a steam trap on the outlet. Water was conducted through suitable piping from the bottom of the heater tank to the annular space between the copper and glass tubes. By throttling with a valve on the outlet end of the apparatus, the water pressure existing in the annular space was varied. It was noticed that when the water was under atmospheric pressure, large bubbles formed on the surface of the copper tube and rapidly left the surface; whereas when the water was at a greater pressure, no bubbles were observed. Photographs taken under these two different operating water pressures are shown in Figs. 8 and 9, the former being for atmospheric water pressure and the latter for the greater pressure.

Of course it is reasonable to expect that inside the heater not all the cold water will flow directly past the tubes; probably a large part of the water is heated by its mixing with heated lower layers of water, and from steam bubbles in the circulation. This probably accounts for the fact that an overall coefficient of heat transfer for a heater is more frequently used rather than individual coefficients. Undoubtedly the water leaving the tube surface will be very close to its boiling temperature with some tendency toward the formation of steam bubbles. It seems reasonable to suppose that the tendency of these bubbles to form and aid circulation will be influenced by the pressure of the surrounding water; the less the pressure surrounding the tubes, the greater is the tendency for the formation of bubbles and the creation of turbulence that would increase heat transfer.

Conclusions

The overall heat transfer coefficients and consequently the capacities of storage water heaters depend to a considerable extent on the water pressure, varying inversely with the pressure.

The capacity was found to remain constant with water pressure variations in the range from 80 lb to 35 lb per square inch gage. Below a pressure of approximately 35 lb per square inch gage, the capacity increases considerably, being highest at the lowest water pressure. This difference due to water pressure is greater at high steam pressures than it is at low steam pressures.

With steam at 20 lb per square inch gage pressure the heat transfer coefficient and the capacity are about 1.8 times as great at atmospheric water pressure as at pressures above 35 lb per square inch gage. With steam at 90 lb per square inch gage the coefficient and capac-

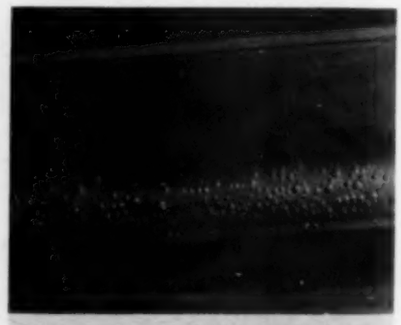

Fig. 8 (*above*)—Photograph of small-scale heat transfer apparatus showing effect of atmospheric water pressure

Fig. 9 (*below*)—Photograph of small-scale heat transfer apparatus showing effect of main water pressure

ity are about 2.7 times as much at the lower water pressures as at the higher water pressures.

The coefficient increases from 310 to 670 Btu per hour per square foot per degree Fahrenheit for the lower water pressure and from 190 to 240 Btu per hour per square foot per degree Fahrenheit for the higher water pressure through the range of logarithmic mean temperature differences of 150 to 230 F.

Cleaning the outer surface of the tubes increased the capacity by 40 per cent at 20 lb per square inch gage steam pressure and 21 per cent at 90 lb steam pressure, with the water under atmospheric pressure.

Visual evidence shows that the higher heat transfer at the lower water pressures is the result of the formation and movement of steam bubbles. These are formed on the tube surfaces, break away and carry heat into the main body of water above the tubes.

For tubes that had been in test service for a short time, these tests indicate a capacity of 30 gal per hour through a range of 40 to 140 F per square foot of heating surface with atmospheric steam pressure and a corresponding capacity of 51 gal at 50 lb per square inch gage. These values compare with 38.5 at atmospheric and 73 gal at 50 lb steam pressure quoted in a storage heater catalog.

Acknowledgment

The results in this paper were obtained in the Department of Mechanical Engineering under the general direction of Prof. G. L. Larson, chairman. Acknowledgment is due J. C. White, power plant engineer of the State of Wisconsin, for his cooperation and also to the following individuals who assisted with the tests: W. F. Gother, E. J. Rosecky, A. J. Thom, E. A. Utecht and R. A. Wilson.

Minnesota Chapter Meetings

February 13, 1939. The February meeting of the Minnesota Chapter was held in the Oak Street Laboratories of the Engineering Experiment Station, University of Minnesota, with dinner served to 110 members and guests. The general arrangements for the meeting were taken care of by Prof. A. B. Algren.

The dinner was followed by a short business session conducted by Pres. J. E. Swenson, after which the various members of the Experiment Station staff were introduced by Professor Algren.

Prof. F. B. Rowley gave a brief outline of the history of the Oak Street Laboratories, their development and possibilities. He also outlined several cooperative research projects now in progress and called attention to the excellent facilities in the laboratories for large scale projects to be conducted in cooperation with industries.

Professor Rowley called attention to the investigations now in progress such as the moisture condensation problem in cooperation with the *National Mineral Wool Association*, the manufacture of a suitable fuel from peat, which is being sponsored by the State of Minnesota, and the manufacture of hydrogen from lignite and the extraction of manganese from manganese ore, both sponsored by the Northwest Research Foundation.

In addition to these projects attention was called to research in fields of concrete under the direction of Prof. C. A. Hughes, and investigations being carried on in connection with internal combustion engines under the direction of Prof. B. J. Robertson.

The general description of the laboratories and their facilities was followed by an inspection of the plant, which included a visit to a large temperature control room, two wind tunnels, another large scale temperature control room for testing insulating materials applied to buildings, the former room being designed for testing the insulating qualities or the air conditioning equipment of railroad cars and truck bodies. The inspection continued on to the four test rooms for complete tests of airplane engines, then to the electric dynamometer equipment for internal combustion engines, the chemical engineering laboratory, and special equipment for the study of the vapor resistance of building materials was inspected, as well as apparatus for measuring the dust in air. A laboratory shop and various pieces of smaller laboratory equipment were also inspected.

January 9, 1939. The January meeting of the Minnesota Chapter was called to order by secretary M. H. Bjerken at the Center for Continuation Study, University of Minnesota, in the absence of Pres. J. E. Swenson and Vice-Pres. F. C. Winterer, with 33 members and guests in attendance.

A brief talk on the seminar at the University and its accomplishments was given by Professor Rowley.

Various committees were called upon to report and the personnel of the 1938-1939 committees was outlined as follows:

Advisory: R. E. Backstrom, with the assistance of all Past Presidents.
Attendance: D. W. Hickey, *Chairman;* and J. R. Hall.
Auditing: C. E. Gausman, *Chairman;* and L. C. Gross.
Entertainment: F. H. Schernbeck, *Chairman;* D. S. Knapp, N. D. Adams and W. F. Uhl.
Legislative: F. B. Rowley, *Chairman;* E. F. Jones and H. M. Betts.
Membership: D. M. Forfar, *Chairman;* C. E. Lund, C. D. Fitts, G. A. Dahlstrom and H. C. Oberg.
Nominating: R. E. Backstrom, *Chairman;* N. D. Adams and C. E. Gausman.
Program: F. C. Winterer, *Chairman;* A. B. Algren, William McNamara, G. S. Bean, R. E. Backstrom and H. S. Morton.
Publicity: F. F. Lange, *Chairman;* and D. B. Anderson.

The guest speaker of the evening, H. C. Rooks, engineer, The Trane Co., La Crosse, Wis., gave a very interesting talk on Evaporative Condensers. In his address Mr. Rooks presented some valuable information to the members and guests.

A motion was then made and carried to hold the regular meetings at the Center for Continuation Study, and the secretary was instructed to make the necessary arrangements.

Professor Algren gave a brief report on plans for the next meeting to be held at the Experimental Engineering Laboratories at the University of Minnesota.

November 28, 1938. This meeting was called to order by President Swenson and turned over to Professor Rowley, who pointed out that this gathering was to take the place of the regular meeting scheduled for December, and was held in conjunction with the seminar conducted by the University of Minnesota and by a committee of members of Minnesota Chapter.

There were 110 members and guests present when F. C. Houghten, director of the Society's Research Laboratory at Pittsburgh, was introduced, along with Prof. A. P. Kratz, University of Illinois, who brought greetings from the Illinois Chapter and who also gave a brief outline of investigations conducted at the Research Residence. L. V. Teesdale, Forest Products Laboratory, was then introduced, followed by Dr. Charles Sheard, Mayo Clinic, who gave a talk on the cooperation between the two divisions of medicine and science. Dr. Keys, University of Minnesota, was presented and spoke briefly concerning his experience with the Society and the medical profession.

N. D. Adams then presented a report on the activities of the National Council and future Society meetings.

J. F. McIntire, vice-president, U. S. Radiator Corp., Detroit, extended greetings from the president of the Society, E. Holt Gurney, Toronto. Mr. McIntire gave a very interesting talk on the various activities of the Society and the need of competent engineers in addition to business men in the Society. Mr. McIntire made the first formal announcement of the F. Paul Anderson award to be presented to Professor Rowley at the Annual Meeting in Pittsburgh.

Skin Temperatures of the Extremities and Effective Temperature

By Charles Sheard, Ph.D.,* Marvin M. D. Williams, Ph.D.** and Bayard T. Horton, M.D.,*** Rochester, Minn.

EFFECTIVE temperature was defined by Houghten and Yaglou[1] as "the temperature and humidity condition, or the heat and moisture condition, of the air which determines the transfer of heat between it and the body." Combinations of temperature, humidity and air movement which induce the same feeling of warmth are designated as thermo-equivalent conditions. From a series of investigations,[2, 3, 4] it was concluded that this scale of thermo-equivalent conditions not only indicates the sensation of warmth but also determines the physiological effects on the body induced by heat or cold.

In a recent report, Hick, Keeton and Glickman[5] concluded that the data which they obtained on the skin temperatures of various regions of the body, of persons very lightly clothed, were most satisfactorily correlated with environmental conditions on the basis of the effective temperature scale or index. Using unclothed subjects, Winslow, Herrington and Gagge[6] found that relative humidity had very little effect on comfort in the zone of body cooling (below 86 F for the unclothed body), whereas the influence of air movement was very great, particularly with low air temperature. The chief physiological response which these investigators found in the zone of body cooling was an adaptive fall in skin temperature.

In this paper the results of investigations concerning the skin temperatures (chiefly of the fingers and toes) are presented, obtained on two subjects, with normal circulation, unclothed, and in the basal metabolic state, under various environmental conditions which ranged from 66 to 77 deg ET as indicated on the psychrometric chart[7] for persons at rest, normally clothed and in still air. The skin temperatures of the fingers and toes have been chosen for the reason that, as pointed out in other papers[8, 9] the extremities play an important rôle in the dissipation of heat from the body and for the further reason that thermal changes of the fingers and toes are the most sensitive or delicate indicators of change in the vasomotor control which, presumably, involves the whole surface of the integument. Furthermore, the changes in the temperatures of the toes are of the greatest significance since, within the range of 66 and 77 deg ET, the temperatures of the forehead, thorax and upper portions of the legs and arms are commensurate and high in value, and the upper extremities (as evidenced by the temperatures of the fingers) are involved but little in the regulation of heat dissipation, whereas definite evidence of vasomotor regulation is demonstrated by the spread of the temperatures of the toes from approximately 22 C to 35.8 C.

Experimental Results

Figs. 1, 2 and 3 show the time-temperature relation-

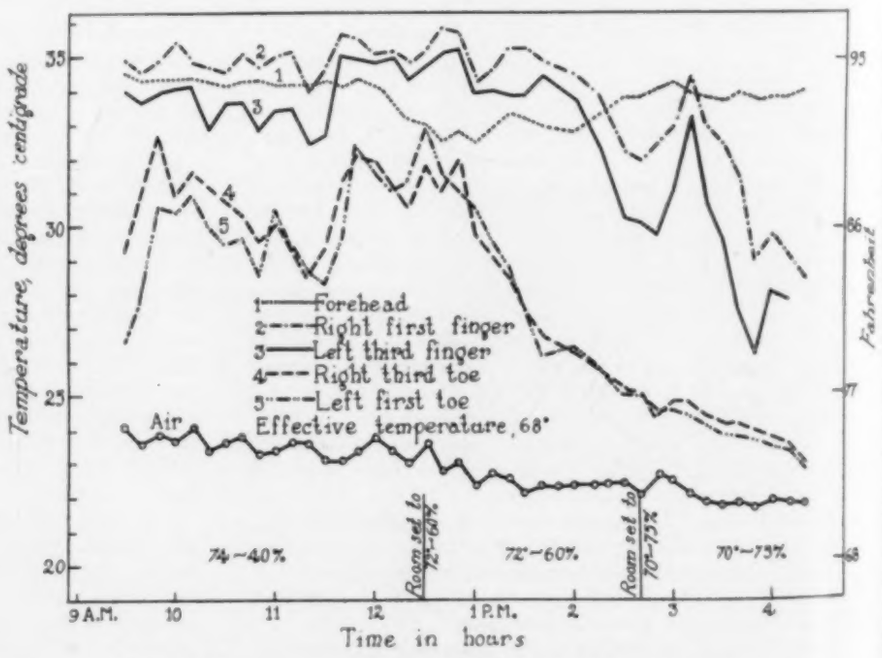

Fig. 1—Skin temperatures of the forehead, fingers and toes of a normal person under an effective temperature of 68 deg

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**Associate, Division of Physics and Biophysical Research, The Mayo Foundation.

***Consultant, Division of Medicine, The Mayo Clinic.

[1]Determination of the Comfort Zone, by F. C. Houghten and C. P. Yaglouglou. (ASHVE TRANSACTIONS, Vol. 29, 1923, p. 361).

[2]Cooling Effect on Human Beings Produced by Various Air Velocities, by F. C. Houghten and C. P. Yaglouglou. (ASHVE TRANSACTIONS, Vol. 30, 1924, p. 193).

[3]Effective Temperature with Clothing, by C. P. Yaglouglou and W. E. Miller. (ASHVE TRANSACTIONS, Vol. 31, 1925, p. 89).

[4]Heat Given Up by the Human Body and Its Effect on Heating and Ventilating Problems, by C. P. Yaglouglou. (ASHVE TRANSACTIONS, Vol. 30, 1924, p. 365).

[5]Physiologic Response of Man to Environmental Temperature, by F. K. Hick, R. W. Keeton and Nathaniel Glickman. (ASHVE JOURNAL SECTION, *Heating, Piping and Air Conditioning*, March, 1938, p. 196).

[6]Relations Between Atmospheric Conditions, Physiological Reactions and Sensations of Pleasantness, by C.-E. A. Winslow, L. P. Herrington and A. P. Gagge. (*Amer. Jour. Hygiene*, 1937, *26*:103).

Presented at the 45th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Pittsburgh, Pa., January, 1939.

[7]Effective Temperature Index of Warmth. (HEATING, VENTILATING, AIR CONDITIONING GUIDE, 1937, *15*, p. 64).

[8]Effects of Changes in Environmental Conditions on Skin Temperatures and the Dissipation of Heat from the Body, by Charles Sheard, M. M. D. Williams and B. T. Horton, (*Amer. Jour. Physiology*, 1937, *119*:403).

[9]Investigations on the Exchanges of Energy Between the Body and Its Environment, by Charles Sheard, M. M. D. Williams and B. T. Horton, (ASHVE JOURNAL SECTION, *Heating, Piping and Air Conditioning*, March, 1937, p. 189).

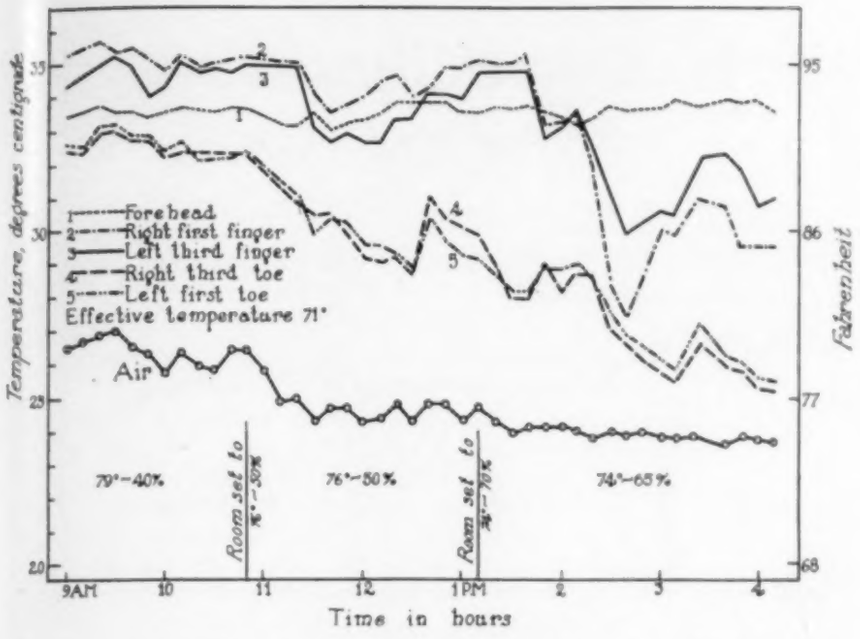

Fig. 2—Curves showing the time-temperature relationships of the forehead, fingers and toes under an effective temperature of 71 deg

ships obtained on the same person when subjected to atmospheric conditions indicated as 68, 71 and 73 deg ET with an air movement of 15 to 40 fpm.

The data obtained at either a low (66 deg) or high (77 deg) effective temperature afford little evidence regarding the equivalence of combinations of dry-bulb and wet-bulb temperatures. The time-temperature relations for the forehead, fingers and toes under an effective temperature of 77 deg show that the skin temperatures of these regions of the body lie within the narrow range of 32.5 to 35.8 C. At 66 deg, the other extreme of the effective temperature scale used in these investigations, the temperatures of the toes remained at room temperature throughout the course of the day, with wide fluctuations in the temperatures of the fingers. Such results would be expected under the extremes of effective temperature which have been cited, for the reason that, at dry-bulb temperatures of 82 to 86 F, there is maximal peripheral dilatation, whereas, at dry-bulb temperatures of 65 to 70 F, there is normal vasoconstriction of the lower extremities with regulation of the dissipation of heat by the upper extremities.

Table 1 contains a résumé of the data regarding physiological responses of two subjects (in the basal state, unclothed, and in the horizontal position) under various effective temperatures. The data show that, within the range of environmental conditions in which the lower extremities play the chief rôle in the regulation of the dissipation of heat, as indicated by the skin temperatures of the toes, there is little effect of considerable changes in the wet-bulb temperatures in contradistinction to the definite influence of relatively small changes in dry-bulb temperature. At an effective temperature of 71 deg, as an illustrative citation, the average temperature of the toes is 26 C (Subject A) when the air temperature is 74 F with a relative humidity of 65 per cent, and 32 C when the air temperature is 79 F with a relative humidity of 35 to 40 per cent.

The question may be raised concerning effects which might be caused by fatigue and minor changes in the production of heat during the course of the day. Figs. 1 to 3 and the data of Table 1 show that three changes were made in the environmental temperature and relative

Table 1—Data Regarding Skin Temperatures of the Fingers and Toes, Blood Pressures and Pulse Rates of Two Normal Individuals Under Various Atmospheric Conditions Indicated by the Effective Temperature Scale

Environmental Conditions			Subject A				Subject B				Comments on Comfort
			Average Temperature, C				Average Temperature, C				
Effective Temperature	Dry-Bulb Temperature, F	Per Cent Humidity	Fingers	Toes	Blood Pressure (mm Hg.)	Pulse Rate	Fingers	Toes	Blood Pressure (mm Hg.)	Pulse Rate	
66	71	40	28.5[a]	22.5	90/60	52	26.5[a]	22.0	94/68	60	Comfortable, except feet cold
	70	55	31.5[a]	22.1	98/64	48	30.0[a]	21.5	94/60	48	Slightly chilly
	69	65	30.2[a]	21.8	88/68	52	29.5[a]	21.7	90/64	50	Slightly chilly, feet cold
68	74	40	34.8	30.5	92/58	56	34.5	29.5	94/60	60	Comfortable
	72	60	30.5	25.6	94/62	54	31.0	23.8	94/66	56	Comfortable, feet cold
	70	75	27.4	23.3	84/68	46	26.5	22.4	94/64	48	Comfortable, feet cold
70	76	40	34.7	29.5	100/74	56	33.5	28.7	96/72	52	Comfortable
	74.5	55	32.0	25.4	88/72	46	31.7	24.4	90/72	50	Fairly comfortable, except feet
	73	65	28.0	23.8	80/60	50	27.4	23.6	90/68	52	Uncomfortably cool
71	79	40	35.1	32.0	96/78	50	35.5	32.5	96/76	66	Very comfortable
	76	50	34.0	29.3	98/70	50	33.8	28.8	98/70	70	Comfortable, slightly cool feet
	74	65	30.5	26.0	90/66	52	32.0	24.5	100/72	60	Uncomfortably cool
73	82	35	35.5	33.9	94/64	51	35.6	34.0	98/70	62	Very comfortable
	79	50	35.0	32.6	90/66	52	35.1	32.7	96/68	64	Very comfortable
	77	70	34.6	29.5	90/60	49	34.3	30.1	96/64	58	Very comfortable, legs and arms slightly moist
75	85	30	35.8	34.6	90/62	51	36.3	33.8	94/60	65	Somewhat uncomfortable
	82	50	35.7	34.3	88/60	64	36.1	34.0	90/62	57	Very comfortable
	79	60	35.7	33.8	92/64	60	35.8	34.6	90/66	52	Comfortable
77	86	40	35.8	34.5	88/66	56	36.0	34.5	104/76	64	Uncomfortable—sweating
	83	60	35.7	34.3	86/64	54	35.7	34.2	106/74	58	Slightly uncomfortable, too warm
	81	73	34.9	32.7	92/70	52	35.1	33.2	102/76	60	Comfortable. Forehead, hands, feet dry

[a]Temperatures of the fingers vacillated considerably, covering a range as great as 6 C. (See Fig. 1.)
Two values of the basal metabolic rate were obtained for each subject, as follows: *A*, —2 (44.5 cal/square meter/hour) and —7 (41.7 cal/square meter/hour); *B*, —10 (39.2 cal/square meter/hour) and —17 (34 cal/square meter/hour).
All data were obtained with the subjects in the basal state, wearing pajamas and lying in the horizontal position.

humidity during a period of seven or eight hours. The order of procedure was to decrease the temperature and increase the relative humidity, for any specified effective temperature, at the end of a two-hour period. Hence the most marked physiological effects, as indicated chiefly by a decrease in the skin temperatures, would occur in all likelihood in the middle to the close of the afternoon and at the time when relatively lower dry-bulb temperatures and higher humidities were maintained. The data of Table 2 indicate that, under an atmospheric temperature of 78 F and 37 per cent (average) relative humidity, there were minor changes in the temperatures of the forehead and fingers of both persons tested, whereas decreases of 1.9 C (Subject A) and 3.4 C (Subject B) were obtained in the temperatures of the toes during the course of the day. The greatest change in the skin temperatures occurred from 2 P. M. to the conclusion of the test. The same order and approximate changes in the daily range of skin temperatures were obtained at 77 F and 60 per cent relative humidity. The application of such correction factors, as are indicated in Table 2, to the data of Table 1 or to the curves of Figs. 1 to 3, does not appreciably affect the results or invalidate the conclusions to be drawn concerning the effects of various thermo-equivalent conditions on the dissipation of heat from the body.

The effects of changes in relative humidity on skin temperatures were investigated further at dry-bulb temperatures ranging from 73 F to 82 F. The two persons who were used in obtaining the data of Table 1 were tested again in a manner which did not involve a predetermined program. During the course of various tests conducted over a period of two months data were accumulated concerning the skin temperatures of various areas of the body when subjected to various environmental temperatures and ranges of relative humidity at any designated dry-bulb reading. The data of Table 3 show that there is little if any effect produced on the skin temperatures by changes in the relative humidity at any designated atmospheric temperature.

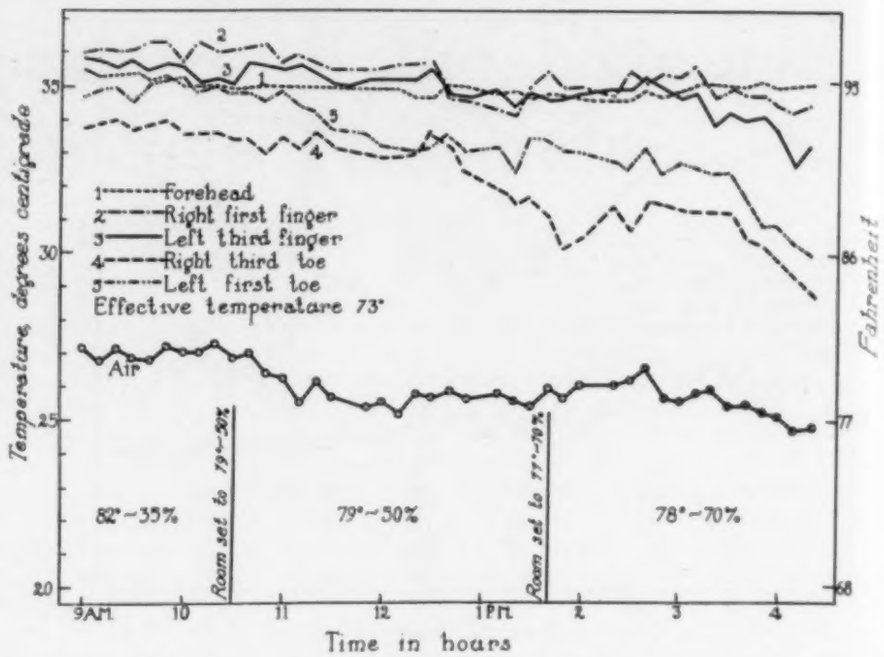

Fig. 3—Skin temperatures of the forehead, fingers and toes of a normal person under an effective temperature of 73 deg

Conclusions

These investigations show that relative humidity has little effect on the skin temperatures of the body, as indicated most sensitively by changes in the temperatures of the fingers and toes, when the body is subjected to various effective temperatures extending from 66 to 77 deg or to fixed environmental temperatures (73 to 82 F) with a fairly wide range (35 to 70 per cent) of relative humidity. It is the authors' conclusion, therefore, that, in the zone of body cooling, the dissipation of heat from the body is dependent chiefly on the environmental temperature and is little influenced by relative humidity, when the person under test is in the basal metabolic state, unclothed and at rest in still air.

Table 2—Chronological Record of the Average Skin Temperatures of the Forehead, Fingers and Toes Obtained on Two Subjects Following an Initial Period of 15 Hours' Fasting. Dry-bulb Temperature 78 F (22.5 C) and 37 Per Cent Relative Humidity

Time of Day	Subject A Average Temperature, C			Subject B Average Temperature, C		
	Forehead	Fingers	Toes	Forehead	Fingers	Toes
9:00—11:30 A.M.	34.5	35.3	31.5	35.1	34.9	32.4
11:30— 3:00 P.M.	34.6	35.6	31.0	35.3	34.7	31.6
2:00— 4:30 P.M.	34.8	34.7	29.6	33.7	34.3	29.0

Table 3—Data Concerning the Effects of Various Environmental Temperatures and Ranges of Relative Humidity at Any Designated Dry-bulb Reading on the Temperatures of the Forehead, Fingers and Toes[a]

Environmental Conditions		Subject A Average Temperature, C			Subject B Average Temperature, C		
Dry-Bulb Temperature	Relative Humidity (Average Value) Per Cent	Forehead	Fingers	Toes	Forehead	Fingers	Toes
73 F (22.8 C)	38	33.8	31.0	24.5	34.7	32.0	23.8
	53	33.5	29.5	23.4	34.5	31.0	23.2
	68	34.2	28.5	23.6	35.2	30.5	23.4
75 F (23.9 C)	41	34.2	35.5	32.5	35.6	34.5	32.0
	54	34.3	34.7	31.6	35.2	35.0	30.8
	64	34.7	33.8	29.6	35.0	33.8	28.4
77 F (25 C)	39	33.6	34.5	29.3	33.5	34.0	29.8
	58	34.1	33.8	28.5	34.5	33.5	27.2
	70	34.5	32.5	27.3	34.8	31.8	26.8
79 F (26.1 C)	32	34.5	35.5	32.2	34.5	34.6	32.5
	50	34.8	35.5	31.5	34.8	34.7	31.6
	64	35.3	34.3	29.8	35.2	35.0	31.1
82 F (27.2 C)	30	34.5	34.6	33.6	35.3	35.0	33.2
	53	35.1	35.0	32.8	35.0	35.5	33.5
	66	35.2	35.0	32.5	35.4	35.4	33.4

[a]Basal metabolic rate (taken at dry-bulb temperature of 77 F and relative humidity 40 per cent); subject *A*, —2 (44.5 cal/square meter/hour); subject *B*, —10 (39.2 cal/square meter/hour).

The data tabulated for any given dry-bulb temperature and range of relative humidities were obtained on the same day on both subjects. The two subjects remained under any given environmental temperature and relative humidity for a period of 2 to 3 hours.

Air Conditioning in Industry (Part 3)

By W. L. Fleisher,* A. E. Stacey, Jr.,**, F. C. Houghten,*** and M. B. Ferderber, M.D.†, (*MEMBERS*)

Test Data and Results on the Physiological Reactions of Individual Workers to High Effective Temperatures

In Fig. 7 are plotted some of the physiological reactions for one of the subjects (D. I.) during a 4-hour test in an air condition of 90 deg ET at 75 per cent relative humidity. This subject was at rest in the control room from 9 to 10 A. M., at which time he entered the test room and worked for 4 hours, or until 2 P. M. Body temperatures and pulse rates were observed every 15 min in the control and a little less often in the test room. The other physiological observations, because of the difficulty incident to making them, were taken at less frequent intervals.

In addition to this subject's (D. I.) physiological reactions as plotted in Fig. 7, his metabolic rate before leaving the control room was 40.8 calories per square meter per hour, and his average systolic and diastolic blood pressures were 104 and 69, respectively.

It will be observed from Fig. 7 that both the body temperature and the pulse rate show a progressive change with time of exposure in the hot atmosphere. This rise in body temperature is logically expected to occur since body temperature must show a proportional rise with accumulated heat. The accumulation of heat is in turn a function of the rate of heat produced in the body and the rate of heat loss from the body to the atmosphere and surroundings. About the only factor which is likely to change this proportional relationship is the slight increase of metabolism with rise in body temperature. This relationship has been credited to the purely chemical law of Van't Hoff, which states that most chemical reactions are a function of and increase with temperature. This is contrary to the best interests of the body which would dictate that with rise in body temperature the metabolic rate, or rate of heat production, should decrease. In other words, the point where metabolism increases with rise in body temperature or with severity of a hot atmosphere represents the initial breakdown of the processes of life, or the adjustment of the internal reactions of the body to meet the external environment. Then it becomes a question of a further small elevation in the severity of the hot condition or length of time of exposure before death occurs. It is interesting to point out that the progression of physiological changes is an accellerative process after the maximum temperature at which the body can adjust its internal reactions to its environment has been reached.

It will be noticed that change in body temperature and also the pulse rate do not follow straight line relationships with time. The body temperature changes rapidly immediately after entering the test room and tends to become a constant with increased length of exposure. This is because the rise in body temperature in the constant atmospheric condition decreases the temperature difference between the body and its environmental air. If a person were able to exist satisfactorily at a sufficiently elevated body temperature, his physiological mechanisms would consequently establish a higher temperature level at which equilibrium could be maintained. This adjustment, however, can only be made within rather small limits compatible with life; and even at lesser elevations in temperature where death or collapse would not occur, the individual ceases to function normally.

Increase in pulse rate is not necessarily a quantitative change resulting directly from exposure, but represents a physiological reaction resulting from nerve stimulus. If psychic factors are eliminated, the stimulus causing an increase in pulse rate is apparently more or less proportional to the rise in body temperature, for increase in pulse rate parallels quite closely rise in body temperature, except in unusual diseased states. It has been shown in other work, however, particularly in the application of fever therapy,[2] that if the patient can be kept

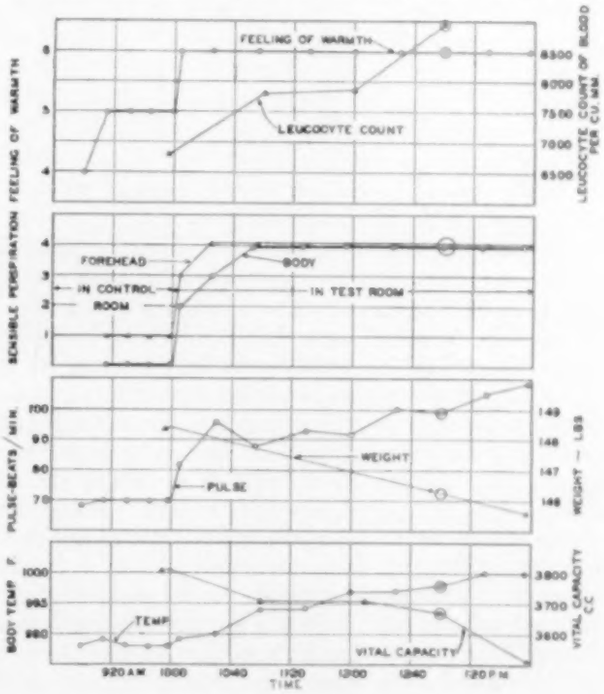

Fig. 7—Physiological response of D. I. at rest in control room and while working in an air condition of 90 deg ET and 75 per cent relative humidity

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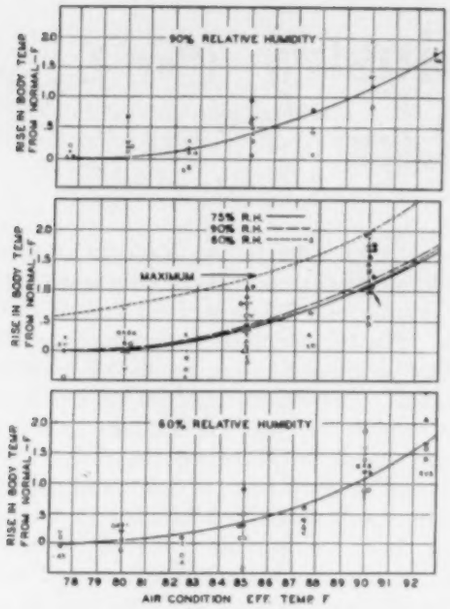

Fig. 8—Relation between rise in body temperature from normal during 3-hour exposure and the effective temperature of the air condition in the test room for 60, 75 and 90 per cent relative humidities. Test points for individual subjects (Table 1) apply to heavy solid curves

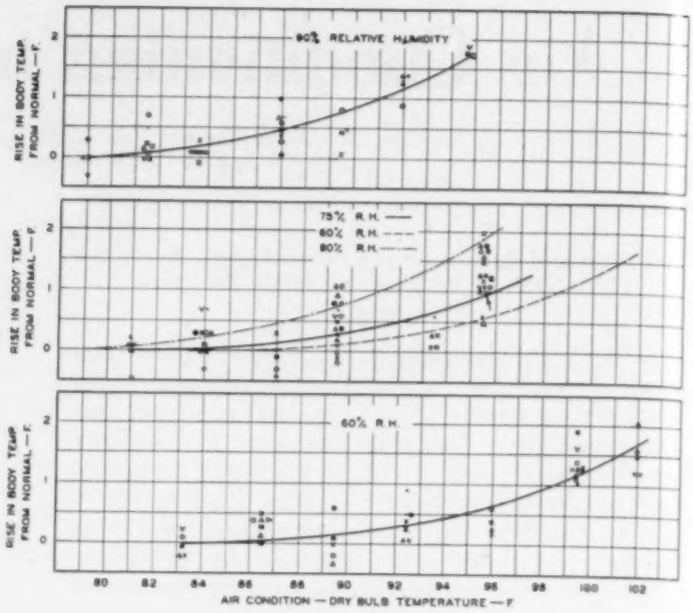

Fig. 9—Relation between rise in body temperature from normal during 3-hour exposure and dry-bulb temperature of air condition in test room for 60, 75 and 90 per cent relative humidities. Test points for individual subjects (Table 1) apply to heavy solid curves

quiet or unexcited, either through his own self control or due to the application of a sedative, a rise in body temperature may not be accompanied by the same increase in pulse rate, except where there is already a high basal metabolic rate or where there may be some physical debility.

Plots or tabulations of the several physiological reactions similar to those in Fig. 7 were made for each individual subject in all of the tests made at 60, 75 and 90 per cent relative humidity. The several reactions after 3 hours of exposure are plotted in Figs. 8 to 21, and the points in each case were taken from the plotted curves similar to those in Fig. 7, or from calculation based upon the rate of change where fewer observations were taken, as in the case of vital capacity and blood pressure. As an example, if the vital capacity was found to be 4000 cc before entering the test room and 3650 cc after 3½ hours of exposure, the hourly rate of change was assumed to be 100 cc, and change in the vital capacity in 3 hours was assumed to be 300.

The relation between the rise in body temperature and the effective temperature, dry-bulb temperature and wet-bulb temperature of the atmosphere for the three humidities studied are plotted in Figs. 8, 9 and 10, respectively. The results of tests at the three different relative humidities are plotted separately. In each case the solid line curves are drawn for the plotted test points for the different subjects (see Table 1) after 3 hours exposure. The resulting curves for 60 per cent and 90 per cent relative humidity tests are superimposed on the 75 per cent relative humidity section of the chart. It will be noted that when temperature rise is plotted against effective temperature there is little separation of the curves for the different relative humidities. In the plot against wet-bulb temperature there is some

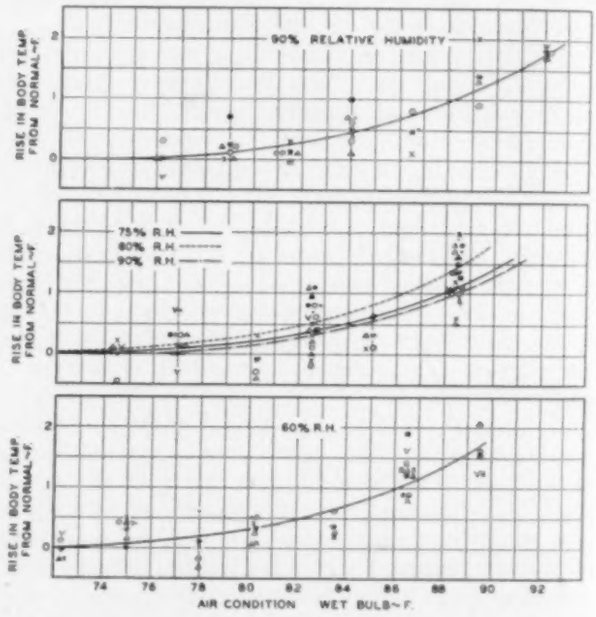

Fig. 10—Relation between rise in body temperature from normal during 3-hour exposure and the wet-bulb temperature of the air condition in the test room for 60, 75 and 90 per cent relative humidities. Test points for individual subjects (Table 1) apply to heavy solid curves

separation while for the dry-bulb plot there is a very significant separation of the three curves. It is of interest to note that when plotted against wet-bulb temperature, the separation is such as to give a series from 60 to 90 per cent relative humidity from the upper curve to the lower, while when plotted against dry-bulb the curves are in the reverse order. This is as should be expected from the relation of effective temperature to wet-bulb and dry-bulb temperatures; that is, for any atmospheric condition giving a certain effective tem-

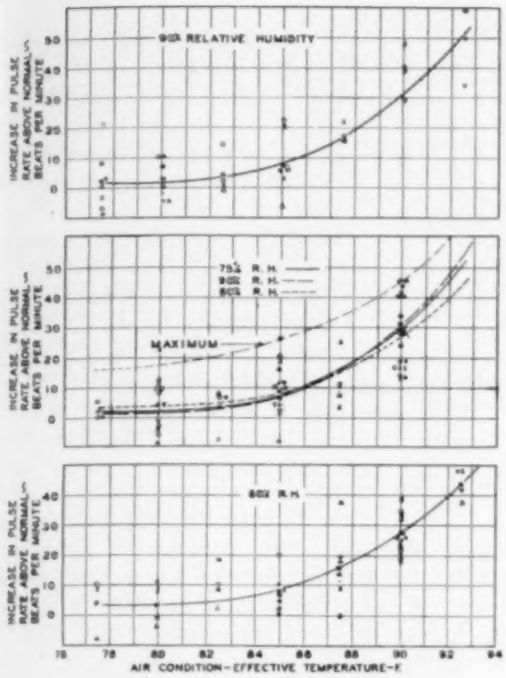

Fig. 11—Relation between increase in pulse rate from normal during 3-hour exposure and the effective temperature of the air condition in test room for 60, 75 and 90 per cent relative humidities. Test points for individual subjects (Table 1) apply to heavy solid curves

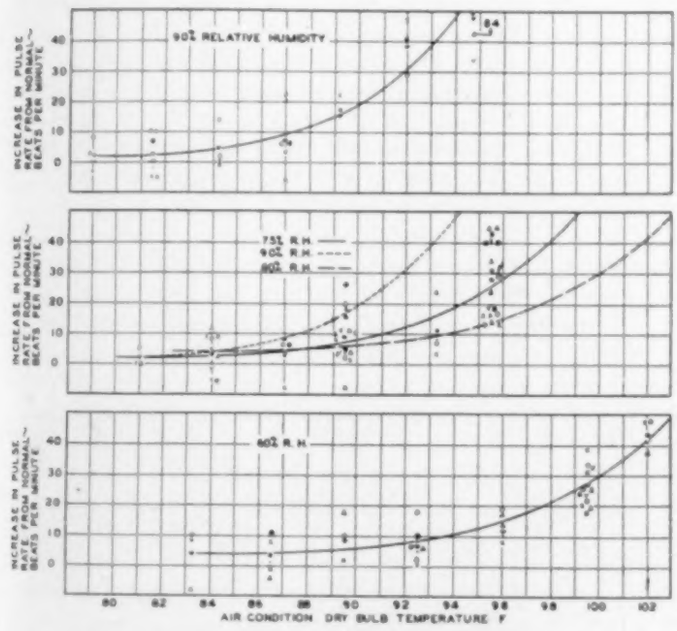

Fig. 12—Relation between increase in pulse rate from normal and the dry-bulb temperature of the air condition in the test room for 60, 75 and 90 per cent relative humidities. Test points for individual subjects (Table 1) apply to heavy solid curves

perature at any relative humidity below 100 per cent the wet-bulb temperature is lower and the dry-bulb temperature is higher than the effective temperature.

The considerable variation for the rise in body temperature of different subjects and for the same subject in different tests in the same air condition indicates that

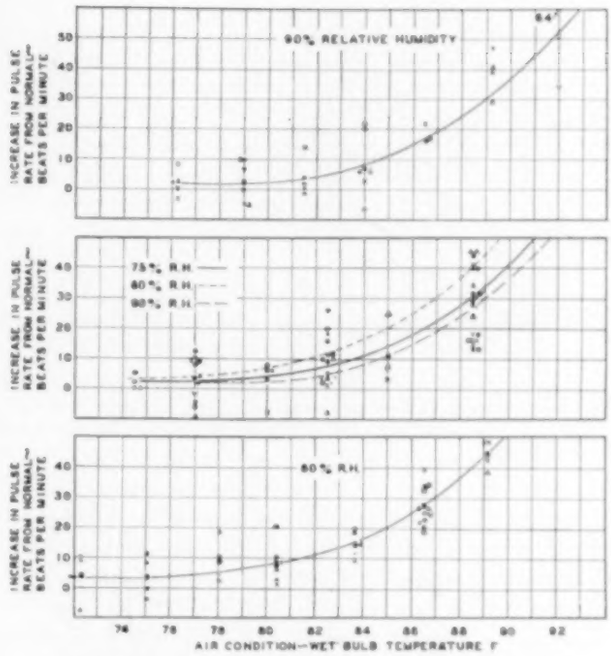

Fig. 13—Relation between increase in pulse rate from normal and the wet-bulb temperature of the air condition in the test room for 60, 75 and 90 per cent relative humidities. Test subjects (Table 1) apply to heavy solid curves

variation in the physiological reactions of individuals or the same individual may be expected at different times. This variation under different test conditions is not much different from the variation in body temperature after ¾ hour of rest in the control room, as given in Table 1.

The increase in pulse rate for all subjects after 3 hours exposure in all tests at 60, 75 and 90 per cent relative humidity is plotted against effective temperature, dry-bulb and wet-bulb temperatures, respectively, in Figs. 11, 12 and 13. In each case the test points for the different subjects fit the solid line curve representing a 3-hour exposure. The curves for 60 and 90 per cent relative humidity are in each case superimposed on that for 75 per cent, in order to make possible a comparison of the effects of tests at the different humidities.

The relation between percentage change in the vital capacity of the subjects and effective temperature of the atmospheric condition for all subjects in all tests in which this observation was made is given in Fig. 14. No discernible variation in the reaction for tests at different relative humidities was indicated, and consequently the points for all three humidities are plotted in this single chart, which shows the changes taking place during a 3-hour period. While there is a rather definite decrease in the vital capacity after 3 hours exposure for the hotter atmospheres, the change is not very significant in that the variation between individuals for the same condition on the same day and for the same individual on different days is greater than the average change due to the severity of the atmospheric condition.

The leucocyte count of the blood in white cells per cubic millimeter for all subjects in all tests in which this observation was made is plotted in Fig. 15. Since this

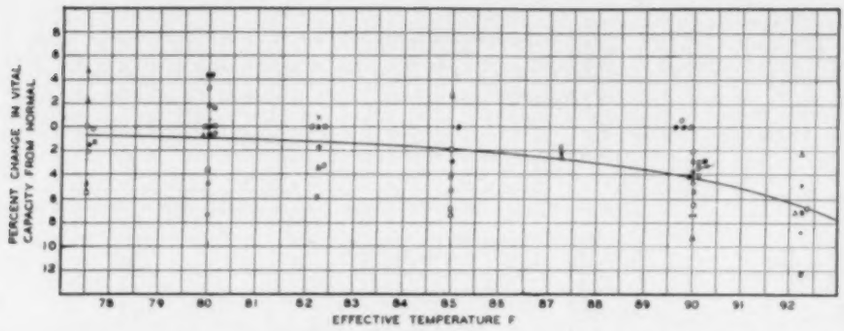

Fig. 14—Per cent change in vital capacity after 3 hours of work in atmospheric conditions of 60, 75 and 90 per cent relative humidities and the effective temperatures indicated. Curve shows change for the three humidities

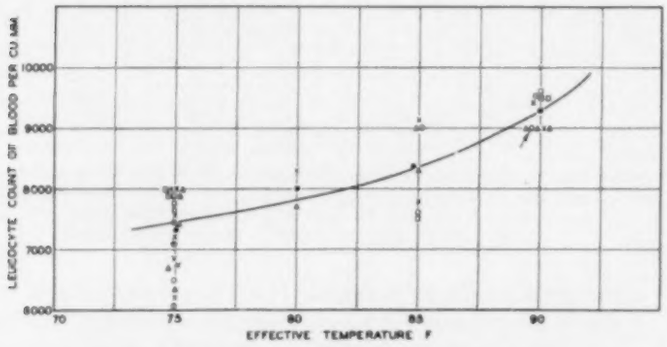

Fig. 15—Change in leucocyte count of blood after 3 hours of work in atmospheric conditions of 60, 75 and 90 per cent relative humidities and the effective temperatures indicated. Curve shows changes for the three humidities

observation was rather difficult to make, in that it required special equipment and skill of manipulation, only a relatively few observations were made. In these tests the count was made in the control room just before entering the test room, and again after 3 hours exposure at work in the hot condition. The observations in the control room were all made on days when the effective temperature maintained was 75 deg. The curve shows a definite increase in the count after 3 hours exposure in the hot conditions. The leucocyte count in the control room is a little higher than is generally given for normal people in normal atmospheres, which fact may be due to its being taken within a couple of hours after the subject's rising in the morning, eating breakfast and coming to work; the latter task frequently involved a walk of some distance. An average of 6000 resulting from variations of from 5000 to 7000 would be more expected from normal, inactive persons in a comfortable atmosphere. The increase in the count after 3 hours of working in the hotter atmosphere is definite and probably of considerable importance. However, unlike elevation in body temperature and increase in pulse rate, no definitely harmful significance can be placed on the increased leucocyte count. But it should be kept in mind that an increase in the count generally accompanies any kind of infection in the body or an elevation in the body temperature. This increase is generally considered as a preparation of the body to cope with infection.

The blood pressure as observed for all subjects in all tests for the three humidities is plotted in Fig. 16. While there is indicated a consistent rise in systolic and a less pronounced fall in the diastolic pressure for exposures in atmospheres above 87 deg ET, these changes in the average curves are not very significant when compared with the wide variation in the values for different individuals on the same day or for the same individual on different days. These curves are, however, consistent with facts observed in the application of fever therapy,[2] where experience shows an increase in the systolic and a decrease in the diastolic pressures with elevation in body temperature due to hot atmospheres. A certain maximum increase in the systolic can occur, after which a further rise in the body temperature may result in a decrease in systolic pressure, which if it continues to fall may result in collapse. Upon complete collapse the systolic pressure may even fall until it approaches the diastolic pressure. For this latter condition of collapse the pulse pressure may be extremely low.

The variations in metabolism with activity and exposure for the different subjects are shown in Fig. 17. The rates of metabolism for basal, seated at rest, and light work are for the subjects in the tests in this study. On mornings when the subject had not eaten breakfast basal observations were made on him after he had lain at complete rest for ¾ hour immediately following arrival at the Laboratory. On other mornings the *seated at rest* observations were made on subjects toward the end of the ¾ hour period in the control room, while the observations for *standing at light work* were made in the test chamber after 3 hours exposure. The points plotted are for tests at the three humidities, there being no significant variation either for tests at different humidities, effective temperatures or lengths of time of exposure. This is in accordance with observations made in earlier work at the Laboratory,[3] in which usually no change in metabolism was found upon exposure to hot atmospheres, excepting that accompanying a considerable elevation in the body temperature. The points giving the metabolic rate at moderate work and heavy work were determined in earlier Laboratory studies.[4] These rates of work consisted of raising and lowering, by means of rope and pulley, a 40-lb weight 7½ and 11 times per minute, respectively, through a distance of 5 ft. Moreover, these rates of work were performed continuously for 3 or 4 hours. The higher rate was designated in the earlier study as *heavy work,* which term is repeated here. Table 2 shows rates of work having as high as 4800 Btu per hour for the resulting metabolism for an average sized man, but it is doubtful if such high rates could be continued for an extended length of time.

The rate of loss in weight of the different subjects while engaged in work under the different atmospheric

conditions is plotted against effective temperature in Fig. 18. Points for all relative humidities are included and these determine the solid line curve. The broken line curves for *at rest* and *heavy work* are from earlier Laboratory studies[1, 5] and will be discussed later in this paper.

Loss in weight of a person in a hot atmosphere, or in fact to a lesser extent in any atmosphere, is due largely to the evaporation of moisture. The major part of this evaporation takes place from the surface of the body, while a lesser portion takes place from the respiratory tract. To a still lesser extent, a continued loss in weight occurs as the result of the exhalation of carbon in the form of carbon dioxide, and this loss is approximately proportional to the metabolic rate. Earlier Laboratory studies[3, 4] have shown that this carbon loss may range from 0.3 oz per hour for a person at rest to 0.8 oz per hour for a person at heavy work. It will be seen, therefore, that it is a rather insignificant part of the weight loss shown.

An analysis of the data indicates a very slight, though not significant, change in the weight loss for different relative humidities at the same effective temperature. As an example, the curves so plotted at 90 deg ET, where the maximum variation occurs, indicate losses of 7, 8½ and 10½ oz per hour at 90, 75 and 60 per cent relative humidities, respectively. While this difference in weight loss with relative humidity is not significant when compared with the wide variation for different individuals on the same day or the same individual on different days, it is nevertheless of interest in considering thermal exchanges between the body and the atmospheric environment.

The relations between perspiration on the body and forehead of the subjects after 3 hours exposure, and the

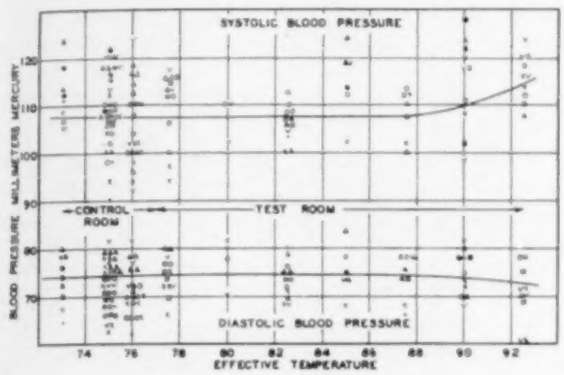

Fig. 16—Change in systolic and diastolic blood pressure after 3 hours of work in atmospheric conditions of 60, 75 and 90 per cent relative humidities and the effective temperatures indicated. Curves show changes for the three humidities

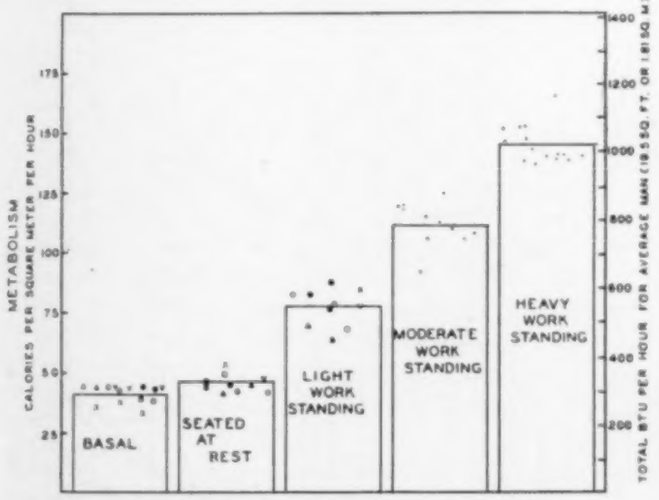
METABOLISM
CALORIES PER SQUARE METER PER HOUR
TOTAL BTU PER HOUR FOR AVERAGE MAN (19.5 SQ. FT. OR 1.81 SQ. M.)
BASAL
SEATED AT REST
LIGHT WORK STANDING
MODERATE WORK STANDING
HEAVY WORK STANDING

Fig. 17—Relation between metabolic rate and degree of activity

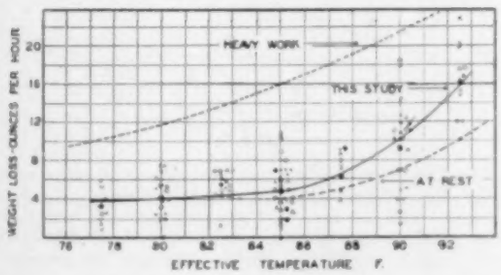

Fig. 18—Relation between weight loss in ounces per hour and the effective temperature of the air condition for the subjects in this study. Curves for persons at rest and heavy work also shown

Table 2—Relation Between Metabolic Rate and Activity[a]

Activity	Hourly Metabolic Rate for Avg. Person or Total Heat Dissipated Btu per Hour	Hourly Sensible Heat Dissipated Btu per Hour	Hourly Latent Heat Dissipated Btu per Hour	Moisture Dissipated per Hour	
				Grains	Pounds
Basal [11]	291	145	145	979	0.140
Seated at Rest [11]	384	225	159	1070	0.153
Reading Aloud (Seated)[17]	420	225	195	1316	0.188
Standing at Rest [11]	431	225	206	1390	0.199
Hand Sewing (Seated)[17]	441	225	216	1458	0.208
Knitting 23 Stitches per Min on Sweater[17]	462	225	237	1600	0.229
Dressing & Undressing[17]	468	225	243	1640	0.234
Tailor[18]	482	225	257	1740	0.248
Singing[17]	486	225	261	1762	0.252
Office Worker Moderately Active[13]	490	225	265	1790	0.256
Light Work Standing[11, b]	549	225	324	2187	0.312
Typewriting Rapidly[17]	558	225	333	2248	0.321
Ironing with 5 lb Iron[17]	570	225	345	2329	0.333
Dishwashing—Plates, Bowls, Cups, Saucers[17]	600	225	375	2530	0.362
Clerk Moderately Active Standing at Counter[13]	600	225	375	2530	0.362
Book Binder[13]	626	225	401	2710	0.387
Shoemaker[13]	661	225	436	2940	0.420
Sweeping Bare Floor 38 Strokes per Min[17]	672	229	443	2990	0.427
Pool Player[13]	680	230	450	3040	0.434
Walking 2 mph [14, 15] Light Dancing [14, 15]	761	250	511	3450	0.493
Light Metal Worker (at Bench)[13]	862	277	585	3950	0.564
Painter of Furniture (at Bench)[13]	876	280	596	4020	0.575
Carpenter[17]	954	307	647	4367	0.624
Restaurant Serving[13]	1000	325	675	4556	0.651
Pulling Weight[11, c]	1041	335	708	4745	0.677
Walking 3 mph[14]	1050	339	711	4750	0.679
Walking 4 mph, Active Dancing, Roller Skating[14, 15]	1390	452	938	6330	0.904
Walking Down Stairs[17]	1444	467	977	6595	0.942
Stone Mason[18]	1490	490	1000	6750	0.964
Bowling[12]	1500	490	1010	6820	0.974
Man Sawing Wood[18]	1800	590	1210	8170	1.167
Swimming[17]	1986				
Running 5.3 mph[17]	2268				
Walking 5 mph[14]	2330				
Walking Very Fast 5.3 mph[17]	2580				
Walking Up Stairs[17]	4365				
Maximum Exertion Different People	3000–4800				

[a] Reference numbers refer to bibliography.
[b] This study.
[c] Pulling 40-lb weight 5 ft over pulley 11 times per minute.

effective temperature of the test room are plotted in Figs. 19 and 20. While there is considerable scattering of points at any given effective temperature for different subjects on the same day and, to a lesser extent, for the same subject on different days, the mass of data shows a very positive increase in the intensity of sensible perspiration with severity of the hot condition. The scattering of the points may in some measure be accounted for by individual variation, and also to the inability of the individual subject to properly evaluate the condition of his perspiration in terms of the index used. This difficulty was greatest when trying to interpret the difference between *clammy, damp,* and *wet* in the case of forehead perspiration. At the same effective temperature there was no distinguishable variation in the degree of perspiration on either the body or forehead with the different relative humidities. Also, there was relatively little variation in the degree of perspiration with time of exposure after the first one-half to one hour of exposure.

The feeling of warmth of the subjects for different atmospheric conditions is plotted in Fig. 21. It is difficult to explain the difference in indicated effective temperature for comfort conditions in this study and previous ones.[6, 7, 8] Whereas a subject in this study felt comfortable at an effective temperature of 82 deg, a person seated *at rest* in former studies required an effective temperature of about 72 deg. Both psychological and purely physical effects come into play in interpreting feeling of comfort, and it is problematical as to the extent to which either may account for the difference. As mentioned previously, the physical factors of larger exposed body surface area and resultant increase in air movement must have some additional effect in increasing the rate of heat transfer. Whether these purely physical changes account for the entire difference as between comfort at 82 deg ET for persons in these tests, and about 72 deg ET for persons seated at rest in previous tests, or whether the psychological interpretation also has its effect is difficult to determine. It is believed, however, that the purely physical factors predominate. In earlier

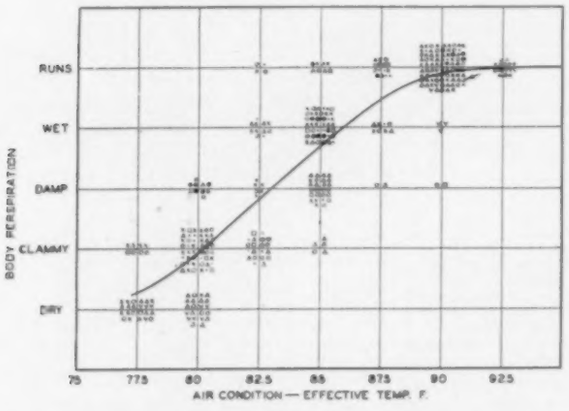

Fig. 19—Relation between degree of perspiration on body after 3 hours exposure and the effective temperature of the air in the test room; for all subjects in tests at 60, 75 and 90 per cent relative humidities

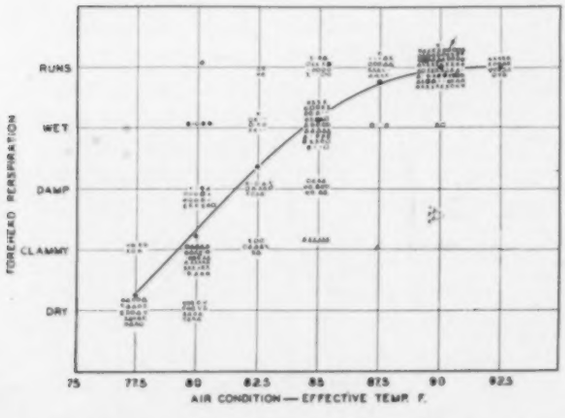

Fig. 20—Relation between degree of perspiration on forehead after 3 hours exposure and the effective temperature of the air in the test room; for all subjects in tests at 60, 75 and 90 per cent relative humidities

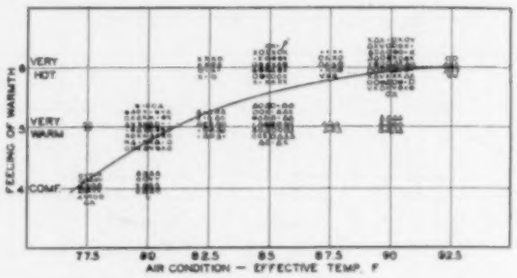

Fig. 21—Relation between feeling of warmth and the effective temperature of the air in the test room for all subjects in tests at 60, 75 and 90 per cent relative humidities

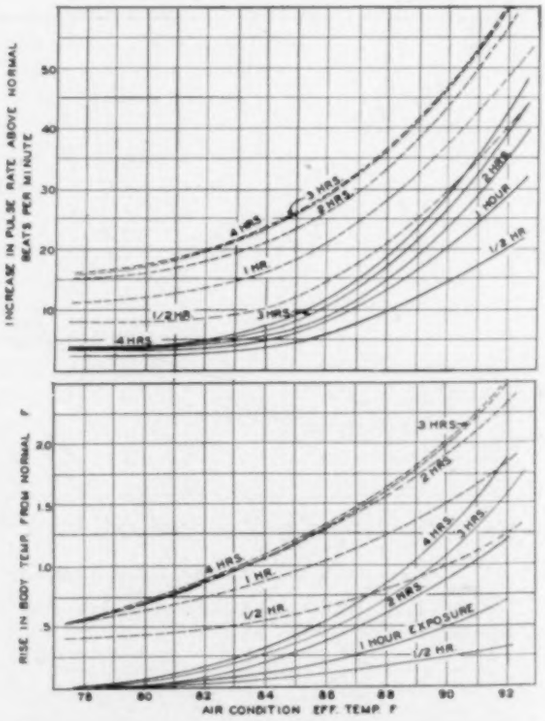

Fig. 22—Relation between the effective temperature of the atmospheric condition and the average and maximum expected rise in body temperature and increase in pulse beats above normal for ½, 1, 2, 3 and 4-hour exposures. Solid line curves represent the average rise in body temperature and average increase in pulse beats of the average subject. Broken line curves represent the maximum expected rise in body temperature and the maximum expected increase in pulse beats for a person physically fit and acclimated for work in hot atmospheres

Laboratory studies[4] it was rather definitely proven in a qualitative, although not in a quantitative manner, that in the same atmosphere sensible heat loss from the body increased for persons at work over that for persons at rest.

The solid-line curves in Fig. 22 show the relation between the effective temperature of the atmospheric condition, for lengths of exposure of ½ hour, 1 hour, 2 hour, 3 hour, and 4 hour, and both the rise in body temperature and the increase in pulse rate. The 3-hour curves are the averages of the curves for the three humidities in Figs. 8 and 11. It will be seen that they form a rather definite series, exemplifying the fact mentioned in connection with Fig. 7, that the rate of rise in body temperature during exposure in a given hot condition decreases with the length of exposure. As an example, the rise during the first ½ hour is greater than the rise during the fourth hour. It should be emphasized in this connection that the indicated rates of rise for 3- and 4-hour exposures in the more severe conditions were based upon the rate of rise during a shorter period; that is, the subjects were not kept in a 92 deg ET atmosphere for 4 hours, but for this condition the time was always limited to 3 hours and frequently to a shorter period.

In discussing the data thus far, consideration has been given almost exclusively to the average reactions of the average subject. While average reactions are the most significant in interpreting what may be expected for the average person for the average exposure, they must be used with caution when determining the worst conditions a person may work under or the length of time a person may work in any hot atmosphere. For this reason the broken-line curves in Fig. 22 were drawn to represent the maximum reactions expected under most circumstances for all workers who have been found to react favorably to hot conditions. This favorable reaction might be brought about either through acclimatization or adjustment in their physical make-up after experience, or through their having a sufficiently adaptable physical make-up. For each length of exposure these curves were drawn so as to be above most of the individual test points for the different subjects, similar to the broken-line curves marked maximum reactions in the middle sections of Figs. 8 and 11.

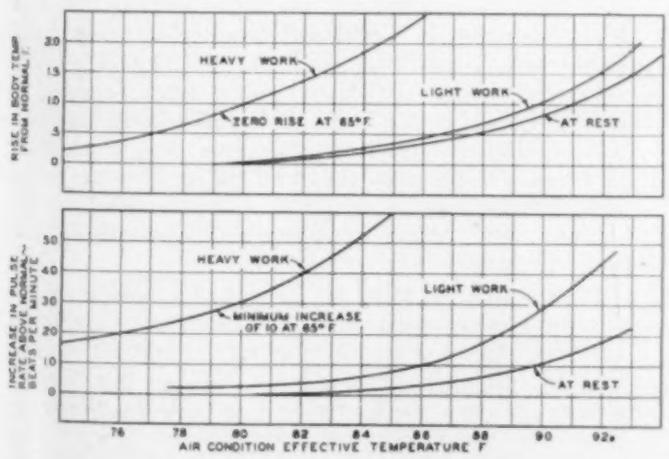
RISE IN BODY TEMP F FROM NORMAL
HEAVY WORK
LIGHT WORK
ZERO RISE AT 65°F
AT REST
INCREASE IN PULSE RATE ABOVE NORMAL BEATS PER MINUTE
HEAVY WORK
LIGHT WORK
MINIMUM INCREASE OF 10 AT 65° F
AT REST
AIR CONDITION EFFECTIVE TEMPERATURE F

Fig. 23—Relation between increase in pulse rate and body temperature from normal after 3 hours exposure and the effective temperature of the air condition in the test room for persons at rest, at light work (this study) and at heavy work

In considering the physical data plotted in Figs. 7 to 21, it is of interest to compare the intensity of physiological reactions, as found in this study, to the similar reactions for persons *seated at rest* and for persons performing *heavy work*, in the earlier Laboratory studies. The metabolic rates for these different conditions have already been compared in Fig. 17, while the weight loss for the three types of activities was presented in Fig. 18. The relation between rise in body temperature and increase in pulse rate, and the effective temperature for the three activities are given for comparison in Fig. 23. The curves for *light work* in this figure were taken as the averages for all three humidities in Figs. 8 and 11 of this study, while the curves for *at rest* and *heavy work* were taken from data published in earlier Laboratory reports.[1, 5] In Fig. 24 are given the relations between sensible perspiration on the body and forehead and the effective temperature of the air condition for persons *at rest, light work*, and *heavy work*. The *light work* curves were taken directly from Figs. 19 and 20 of this report, while the data for *at rest* and *heavy work* were again taken from earlier Laboratory reports.[1, 5]

It is hard to account for the practically equal rise in body temperature and the greater degree of sensible perspiration at a given effective temperature for persons seated *at rest* when compared with persons standing and performing *light work* as in this study. Since these former data were collected several years ago under different conditions, it is possible that the interpretation of sensible perspiration might have differed slightly. On the other hand, these facts may be accounted for by the purely physical factor of increased rate of heat dissipation for a person standing up and moving about, as discussed previously.

The purpose of this study is to supply information which may be used in the guidance of labor, employers, and others interested in the economic and social aspects of working conditions in determining atmospheric conditions under which a person may work a full day or shorter periods without harmful effects. The curves for rise in body temperature and increase in pulse rate for different times of exposure, shown in Fig. 22, offer a ready picture to enable one to determine the relation between time of exposure and the effective temperature, for these physiological reactions.

This study does not in itself offer conclusive information on the severity of a physiological reaction that a

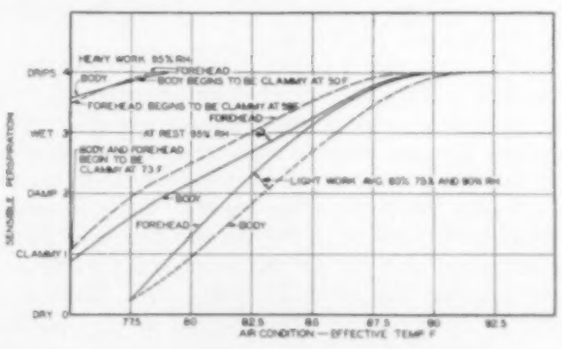

Fig. 24—Relation between sensible perspiration on the body and forehead and the effective temperature of the air condition in the test room for persons at rest, light work (this study) and heavy work

Table 3—Physiological Changes and Conditions Accompanying a Maximum Expected Rise in Body Temperature of 1½ Deg and 1 Deg in from ½ to 4 Hours Exposure. Also Similar Data Accompanying a ½ Deg Average Rise in Body Temperature in Equal Exposures

LENGTH OF EXPOSURE	E.T.-F	MAX. EXPECTED		AVERAGE		VITAL CAPACITY		LEUCOCYTE COUNT		WEIGHT LOSS		PERSPIRATION				COMFORT	
		BODY TEMP. RISE-F	PULSE INCREASE BEATS/MIN	BODY TEMP. RISE F	PULSE INCREASE BEATS/MIN	AVG. % CHANGE	MAX. EXPECT. % CHG.	AVG. PER CU. MM	MAX. PER CU. MM	CHANGE OZ./HR. AVG.	CHANGE OZ./HR. MAX.	BODY AVG.	BODY MAX.	FOREHEAD AVG.	FOREHEAD MAX.	AVG.	MAX.
1/2	93.5	1.5	49	0.38	25												
1	90.0	1.5	37	0.50	20												
2	87.6	1.5	32	0.55	13												
3	87.4	1.5	34	0.62	15	-2.8	-8.2	8700	9600	6.3	13	WET	RUNS	RUNS	RUNS	VERY WARM	VERY WARM
4	87.2	1.5	33	0.70	15												
1/2	90.0	1.0	30	0.25	14												
1	85.6	1.0	20	0.18	7												
2	83.4	1.0	20	0.16	6												
3	83.3	1.0	22	0.21	6	-1.5	-6.5	8200	8900	4.5	9	DAMP	WET	WET	RUNS	COMF	VERY WARM
4	83.1	1.0	22	0.23	6												
1/2	95.0	1.75	57	0.50	30												
1	90.0	1.50	37	0.50	20												
2	87.1	1.43	30	0.50	12												
3	86.4	1.35	30	0.50	11	-2.3	-8.0	8600	9400	5.5	11.5	WET	RUNS	RUNS	RUNS	VERY WARM	VERY HOT
4	85.6	1.27	28	0.50	11												

worker may be expected to endure without harmful effects on his health, or without undue discomfort. Comparison of the data plotted in Figs. 8 and 11 indicates that an increase in pulse rate of about 24 beats per minute accompanies a rise in body temperature of about 1 deg, for moderate conditions of exposure. For more severe conditions the pulse rate increases proportionately at a higher rate for the same rise in body temperature. Rises in body temperature of ½, 1, 1½ and 2 deg are, respectively, accompanied by increases in pulse rate of 12, 25, 38 and 70. The fact that rise in body temperature and increase in pulse rate always appear to be a positive index of exposure to hot conditions makes it seem most desirable to base any limitation, either on length of exposure or severity of a hot condition in which a man should work, on these two factors. As pointed out previously, rise in body temperature represents a more positive mathematical relationship to the atmospheric condition, and therefore, it should be accepted as the better index of the two.

While this study was not designed particularly to determine the allowable rise in body temperature of workers in industry, the observations of the effect of temperature rise on subjects in this study as observed by the investigators and the medical men in attendance offer some basis for a limitation. These observations have led the authors to recommend that the limitations be so placed that no worker be expected to have more than 1½ deg rise in body temperature. Table 3 gives some of the maximum expected and average physiological reactions which would accompany a 1½ deg maximum rise in body temperature. A lack of proportionality between the maximum and average rise in body temperature is apparent and may be due to greater errors in locating the maximum curves from test points. It will be noted that the different lengths of exposure in the several effective temperature conditions all give an approximate average rise in body temperature of ½ deg. If a ½ deg average rise were accepted instead of a 1½ maximum rise, atmospheric conditions of 95, 90, 87.1, 85.6 and 86.4 deg ET could be endured for ½, 1, 2, 3 and 4 hours, respectively.

The major criticism of this investigation as outlined and performed has been that the study would give data only which might be used in limiting hot atmospheric conditions for one type of physical activity, and further, that this particular activity does not accurately simulate any practical work in any industry. A valid answer to this criticism is found in the often demonstrated fact that metabolism is a true and acceptable (in fact, the only true and acceptable) index of intensity of physical activity or work. Hence, all types of work resulting in a given metabolic rate in the same atmospheric condition will undoubtedly result in the same heat elimination problems and therefore the same rise in body temperature, increased pulse rate, and other physiological reactions resulting from interference with heat dissipation from the body.

If this point of view can be accepted, and it is believed that it can, then the relation between physiological reactions and the length of exposure to the effective temperature of the air condition for a relatively few rates of physical activity should make it possible to determine these relationships for any other rates of work. The establishment of such relationships is attempted in Fig. 25, where rise in body temperature and increase in pulse rate for various lengths of exposure are plotted against the metabolic rate. This study offers rather complete

information for these relationships for *light work*, accompanied by a metabolic rate of 76 cal per square meter per hour, and earlier Laboratory studies[4] offer a limited amount of information for persons *at rest* and at *heavy work*. Based upon the average rise in body temperature and increase in pulse rate during a 3-hour period, the same solid-line curves are drawn through well-established points for the three metabolic rates. Maximum expected rates of rise in body temperature and increase in pulse rate for a 3-hour period are available both from this study and the earlier investigation for persons *at rest*, and it was these data and the position of the solid-line curves on this figure that were the determining factors in locating the broken-line curves.

While the meager amount of data available for plotting the curves in Fig. 25 does not establish too accurately their exact position between *light work* and *heavy work*, particularly when the maximum physiological reactions are considered, it does indicate that with relatively little added information curves acceptable for practical use could readily be established.

In order to supply sufficient information for accurately plotting curves similar to those in Fig. 25, another series of tests should be made with the subjects performing *heavy work*, so as to establish with greater confidence these latter points. In addition, a somewhat less comprehensive study at a metabolic rate somewhere in the neighborhood of 100 cal per square meter per hour should aid in fixing the curvature and exact location of the curves between *light work* and *heavy work*. This seems particularly desirable because of the apparent rapid change in the direction of these curves in the neighborhood of *light work*. Moreover, since this study was limited to high humidities, a few additional tests should be made at lower humidities.

With sufficient additional data and the development of perfected charts similar to Fig. 25, one should be able to determine with confidence the temperature rise and the pulse increase for any period of exposure in any hot atmosphere for all rates of work for which the metabolism is known. While the amount of work incident to a continuation of this study is rather extensive, it is quite insignificant when compared with the consequential effects it should have in the application of air conditioning to industry. The cost should be insignificant when compared with the economic saving in health, efficiency, comfort, and the satisfaction accruing to labor, industry and the entire social system.

It is of interest to consider a hypothetical application of the information contained in Fig. 25, to an unstudied type of employment, assuming that no change in the character of the curves in Fig. 25 would result from further study. A few workers performing the new type of work in question would be given a metabolism determination while working. Assume that they show a metabolic rate of 100 cal per square meter per hour, and that the limitation of 1½ deg maximum rise in body temperature suggested is to govern the case. The broken-line curve, 1½ *deg max 3-hour*, will apply, which shows that the worker can engage in this new activity for 3 hours at about 85 deg ET. Similar charts supplied for ½-, 2- and 4-hour exposures would make it as simple to determine the severity of the atmospheric condition which would give a maximum expected elevation of body temperature of 1½ deg in these lengths of exposure.

In order to make available adequate data for estimating the minimum air conditioning requirements for hot industries, future work of the Committee should therefore include:

1. Additional tests with the same type of *light work* in the same atmospheric conditions during the winter season, in order to determine whether men working under these conditions react the same in winter and summer.

2. A series of tests with the same type of *light work* at a lower relative humidity, for example 30 per cent, in order to more adequately determine whether effective temperature alone need be considered, when a wider range of relative humidities is encountered.

3. A series of tests with *heavy work*, in order to supply adequate data for work giving a metabolic rate of 145 cal per square meter per hour, so as to more satisfactorily locate the curves for average and maximum reactions as drawn in Fig. 25.

4. A series of tests with a rate of work giving a metabolic rate of about 100 cal per square meter per hour, in order to better establish the curves drawn in Fig. 25, between *light* and *heavy work*.

5. A series of tests at rest, collecting data under precisely the same atmospheric conditions as in this study, in order to verify or correct the apparent conclusion from this study that a person standing up and performing *light work* has about the same rise in body temperature, a somewhat less degree of perspiration, and

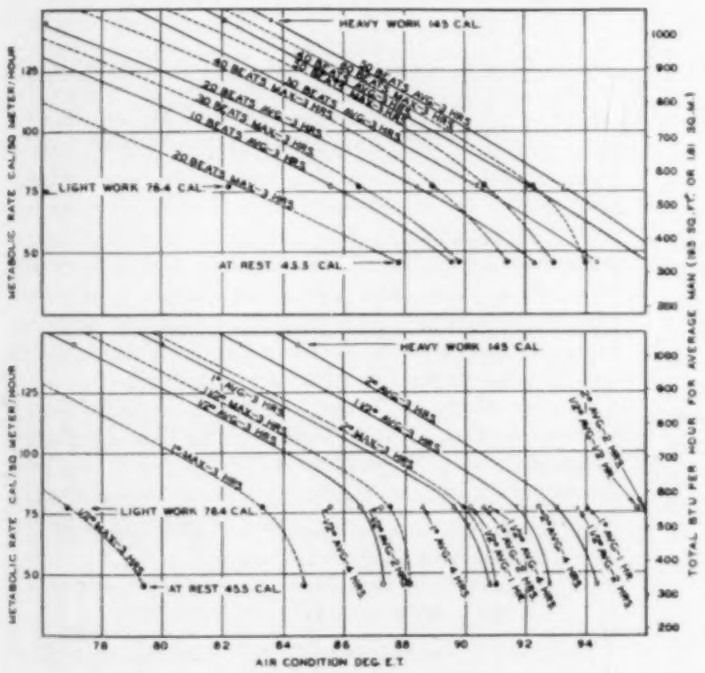
HEAVY WORK 145 CAL.
LIGHT WORK 76.4 CAL.
AT REST 45.5 CAL.
METABOLIC RATE CAL/SQ METER/HOUR
TOTAL BTU PER HOUR FOR AVERAGE MAN (183 SQ. FT. OR 18.1 SQ. M.)
AIR CONDITION DEG. E.T.

Fig. 25—Relation between the effective temperature of the atmospheric condition and the metabolic rate for rise in temperature of ½, 1, 1½ and 2 deg, and for increase in pulse beat of 10, 20, 30, 40 and 50 beats per minute. Solid line curves represent the average rise in body temperature and the average increase in pulse beats of the average subject. Broken line curves represent the maximum expected rise in body temperature and maximum expected increase in pulse beats for a person physically fit and acclimated for work in hot atmospheres. Maximum curves drawn through points indicating metabolic rates obtained from maximum at rest and light work data are drawn similarly to the curves for the average reaction. Points indicating the position of the curves for the several rises in body temperature in from ½ to 4 hours given for light work only

less feeling of warmth than a person seated at rest.

6. Additional work to determine the effect of air movement in reducing the severity of the physiological reactions for different types of work at the same dry-bulb temperature and relative humidity. In this connection, work already carried on at the Laboratory,[5,9,10] has established the effect of air movement on the comfort and presumably the physiological reactions of persons at rest, while a smaller amount of data has been reported for persons working. This new phase of the study should be so planned as to determine the extent to which these earlier findings may apply.

Codes

Within the next few years, there may be anticipated the enactment of many laws and regulations of a social nature. No doubt many of these will have for their subject air hygiene as affecting the health of the worker. This work under the Committee on Air Conditioning in Industry should offer a true foundation for these codes and standards. It may be assumed that regulatory bodies are responsible only for the health of the workers and are therefore interested in those conditions of the atmosphere which adversely affect it. As an example for the need of an understanding and knowledge for those conditions of temperature and humidity which may affect the health of the worker, the State of Massachusetts has had for several years regulations limiting temperatures and relative humidities in textile factories. There can be no doubt that those responsible for these regulations were moved only for the best interests of the worker and that the limiting atmospheric conditions written into the regulations were for his protection.

The maximum dry-bulb temperatures and absolute humidities specified in the Regulations, interpreted as effective temperatures, range from a maximum of 89.3 deg ET to a minimum of 59.3 deg ET. Assuming for the moment that the work of this summer was more than of an exploratory nature and that the critical effective temperature for health occurred when there was a maximum expected increase for any worker of 1.5 F in body temperature with an increase of pulse of 28 beats per minute; or an average increase for the group of 0.5 F in body temperature and 12 pulse beats per minute, then the conditions laid down in the Regulations show a need for fundamental data. All those specified conditions above 91 F dry-bulb are beyond the critical temperature as found for this intensity of activity, so the purpose of the Regulations is not fulfilled as far as the health of the worker is concerned. Those temperature and humidity conditions below 91 F in the Regulations have no effect on the health of the worker and are only a manufacturing limitation. This code was copied from the English Act of 1911 which has been revised in 1935. The revised English Act limits the operation of the mill to times when the wet-bulb temperature does not exceed 80 F.

Codifying critical effective temperatures for industries of the textile mill type is rather simple when compared to those which have conditions above the critical effective temperature during a large portion of the time. If the same criteria of health condition is maintained, it may be necessary and advisable to make provision in the codes for a physical examination of the worker, thus eliminating the unfit and also to provide a limitation of the duration of continuous activity between rest periods. The temperature and humidity found in deep mines offer an excellent example of these extreme conditions. The general sources of heat in the mines are wall surfaces, men, explosives and the cooling of freshly opened rock. The heat from the freshly opened rock is by far the most important. Many of the deep mines are naturally wet and added to this natural moisture in those where the rock is of a siliceous nature, water is used to settle the dust with a consequent high relative humidity. Saturated temperatures of 90 to 95 F are frequently encountered. In the mining district of South Africa, a temperature of 93 F saturated is considered the critical above which many cases of heat prostration occur.

By government regulation, 30 cfm of air must be supplied per person employed underground throughout the 24 hours. Working periods are short, interspersed with rest periods. In other places, many of the deep mines are dry with a much higher rock temperature. The resulting effective temperature is, however, approximately the same and the problem of protecting the health of the worker remains.

When codes are written for industrial conditions of this severity, the laboratory findings of the physiological effects on the worker of these intense conditions of temperature and humidity will be of great value and will supply necessary fundamental information on which to base a code protecting the health of the worker and benefiting the employer through a smaller labor turnover and a greater efficiency of labor.

Engineering

This research will, when completed, determine the critical effective temperature for persons engaged in various degrees of activity. It is expected that as a result of this research tabular information may be compiled giving the approximate critical effective temperatures corresponding to different activities within certain predetermined limits of rise in body temperature and pulse rate. These data will act as a guide to the engineer in the selection of air handling equipment necessary to produce atmospheric conditions below the established critical temperatures for different industries.

Because of the wide range of metabolic activities for different types of work it is conceivable that there might be three different general types of equipment used for producing the desired results which may be classified:

1*a*. Ventilating equipment handling untreated outside air.
b. Open windows in side walls and roof.
2. Air conditioning equipment with evaporative cooling.
3. Air conditioning equipment with refrigeration.

Each one of these classes has very definite limitations which depend not only on the activity or rate of metabolism of the persons engaged but also on the atmospheric conditions met with in the different localities. It can be readily seen that critical temperatures approaching even the higher brackets may be found in subtropical climates and it is also true that in some localities in the north and located on a body of cold water, conditions of quite low critical temperatures could be met without any equipment or that handling only untreated outside air.

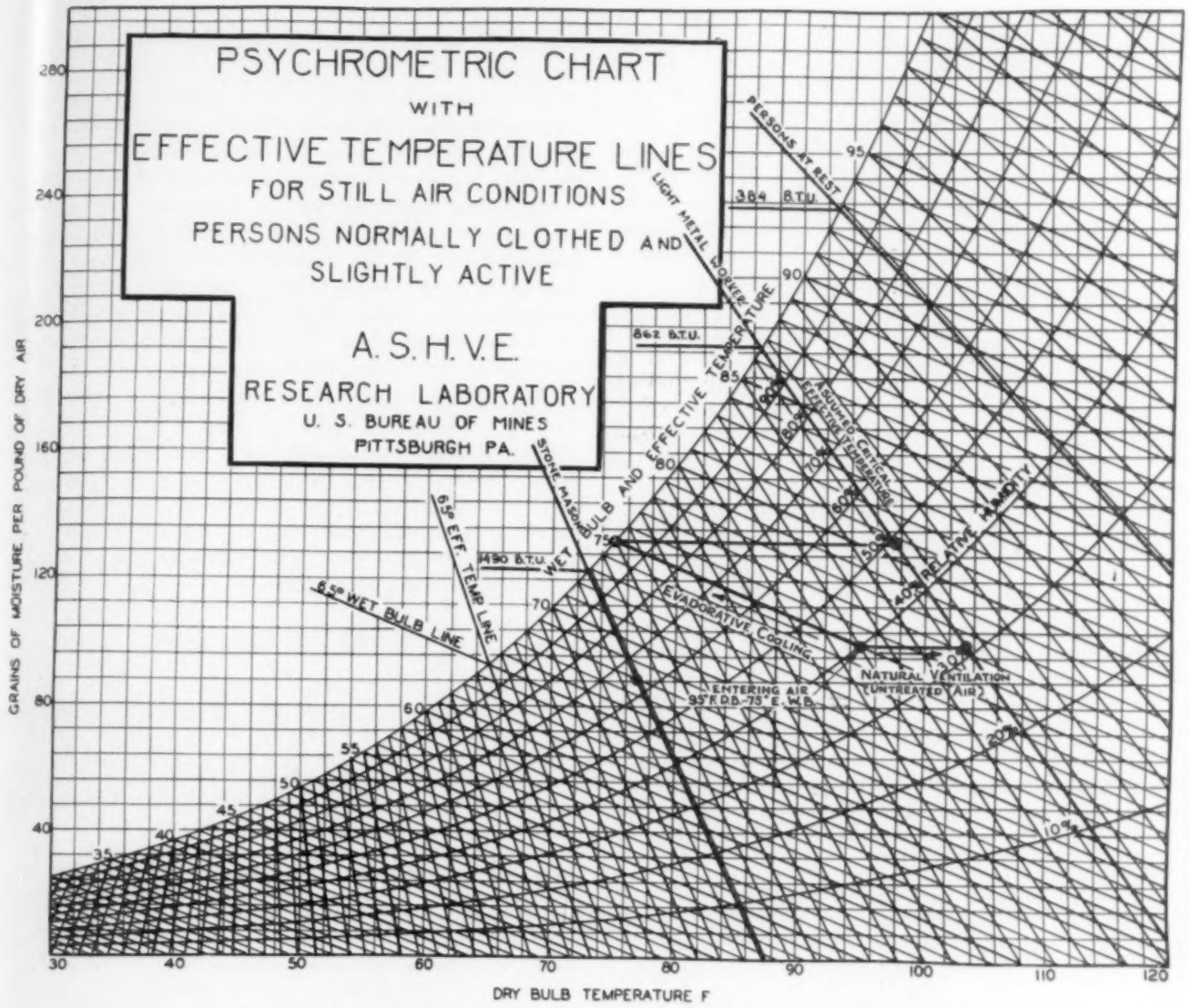

Fig. 26—Psychrometric chart showing critical effective temperatures for three activities and air conditioning processes to produce desired results

In Fig. 26 will be noted three critical effective temperatures corresponding to a 1½ F body temperature rise that have been selected from data reported in this paper and which are enumerated herewith:

	Critical Effective Temp Deg	Metabolic Rate Btu per Hour
1. Persons at rest	92.8	384
2. Light Metal Worker	86.5	862
3. Stone Mason	72.8	1490

Also on this chart is plotted the maximum outside air condition of 95 F dry-bulb and 75 F wet-bulb which is assumed in this case to be applied for illustration to the critical effective temperature line of 86.5 deg corresponding to a metabolic rate of 862 Btu per hour of a light metal worker.

With these assumed conditions it would be necessary for equipment handling untreated outside air to have a capacity in cubic feet of air per minute sufficient to absorb the sensible heat in the room with a temperature rise of 8.8 F which would result in a final condition of the work space of 103.8 F dry-bulb and 77.3 F wet-bulb as may be referred to in Fig. 26.

In those cases where evaporative cooling may be used the amount of cooling depends not only on the outside wet-bulb temperature, but also on the maximum humidity which may be permitted within the enclosure. The amount of air which would be handled by an evaporative cooling device varies directly with the cooling to be accomplished. When evaporative cooling is used for the outside air conditions as previously cited the capacity of the equipment in cubic feet of air per minute would be based on a temperature rise of 22.8 F instead of 8.8 F when using untreated outside air to obtain the same effective temperature. The resulting condition in the space would be 97.8 F dry-bulb and 80.7 F wet-bulb. Thus it is evident that this type of equipment could be used to produce the conditions below the desired critical effective temperature for the degree of activity assumed. For other outside wet-bulb temperatures the limitations of this type of equipment may be readily visualized on the chart. As the wet-bulb temperature is reduced, the effective range of temperature of the air introduced is increased which results in smaller air conditioning equipment and the reverse is true with higher initial outside

wet-bulb temperatures. It may generally be assumed that with evaporative cooling the effective temperature varies inversely with the relative humidity. For example, if it were possible to maintain within an enclosure 80 per cent relative humidity there would be a temperature rise in the air from a saturation point of 7.5 F dry-bulb. However, if it were possible to maintain 70 per cent in the same enclosure the temperature rise of the air would be 11.5 F so that the amount of air required to absorb any specified amount of heat would be proportionately reduced. Unfortunately, the maximum cooling to be obtained by this method is limited by the outside wet-bulb temperature. If a locality had an outside wet-bulb of 75 F and it were possible to carry 70 per cent relative humidity within a plant, then the inside temperature of the plant would be 86 F which is, of course, based upon providing sufficient air to absorb the heat load produced within the plant. These conditions correspond to an effective temperature of 81.2 deg.

In many industries, 65 per cent relative humidity is considered the maximum allowable due to danger from corrosion, etc. With 65 per cent humidity, there is a temperature difference between the saturation point and the dry-bulb temperature of 13.5 F, so that there would result a final dry-bulb temperature of 88.5 F corresponding to an effective temperature of 82.5 F.

As the maximum outside wet-bulb temperature is decreased, so the indoor conditions can be reduced and the reverse is also true. For example, in the south, where it is not unusual to have wet-bulb temperatures of 81 F, then with 65 per cent relative humidity there would be a dry-bulb of 94.5 F corresponding to an effective temperature of 87.3.

To maintain lower critical temperatures it will be necessary to use refrigeration in some form. In the northern part of the country well water of sufficiently low temperature may be obtained. However, in many places it would be necessary to make use of artificial refrigeration. When meeting this condition it might be better from an economic standpoint to shorten the hours of labor or intersperse the periods of activity with intervals of rest so that the same physiological reactions on those persons engaged in this type of work would remain within the healthful limits.

For those degrees of activity where the critical effective temperature is below the outside wet-bulb then some form of refrigeration will be necessary. The capacity of the equipment to fulfill these conditions will depend upon the relative humidity to be maintained within the work spaces. The size of the refrigerating equipment will be smaller as the per cent of relative humidity allowed in the work space is increased, and the amount of air to be handled by the equipment will increase directly with the relative humidity. Assuming that the critical effective temperature for a stone mason is 72.8 deg and that the outside temperature condition is 95 F dry-bulb and 75 F wet-bulb and that it is desired to maintain 65 per cent relative humidity in the work room, it will be necessary to introduce air having a dew-point of 64 F. Therefore, all the air handled by the air conditioning equipment would have to be cooled to 64 F and the amount of air to absorb the heat in the work space would be based on a temperature rise of 13 F which would result in a final condition in the work space of 77 F dry-bulb and 68 F wet-bulb which is equivalent to an effective temperature of 73 deg. The refrigerating capacity necessary to accomplish these results would vary with the requirements for ventilation. When exhaust systems for removing the dust from the cutting surfaces are such that all outside air is necessary, then the capacity of the refrigerating equipment will be sufficient to lower the total amount of air from 95 F dry-bulb and 75 F wet-bulb to 64 F. If the amount of air which is supplied to maintain the desired critical effective temperature is more than that exhausted, then a portion of the air may be returned to the equipment at room conditions with a decrease in the refrigerating requirements.

Engineering calculations to determine the capacity of the air conditioning equipment may be based on procedure outlined in the HEATING, VENTILATING, AIR CONDITIONING GUIDE, 1938 (Chapters 5, 6, 7 and 8). All sources of heat in the space to be cooled must be carefully surveyed. After the proper amount of air has been determined, it should be distributed in the same pattern as the heat load occurs.

Summary and Conclusions

1. The paper reviews the developments of present understanding, or lack of understanding, of the relation between length of exposures of workers to hot atmospheres and the resultant disturbance of their physiological equilibrium.

2. The relations are given between lengths of exposure in hot atmospheric conditions of different effective temperatures and changes in various physiological reactions, such as body temperature, pulse rate, vital capacity, leucocyte count, weight loss, forehead and body perspiration, and feeling of warmth. Body temperature and pulse reactions are indicated as being the most significant reactions correlated with the severity of the hot condition. Moreover, since rise in body temperature is shown to be an adequate index of disturbed physiological equilibrium, it is recommended for use in limiting the length of exposure of workers in hot atmospheres. An average body temperature rise of ½ deg in the average worker qualified by physical make-up and experience to work in hot industries, or a 1½ deg rise in an occasional worker, is suggested as the limiting reaction. Also given are other physiological derangements accompanying a ½ deg rise in the body temperature of the average worker, or a 1½ deg maximum expected rise in the temperature of the occasional worker.

3. A chart is presented for indicating the rise in temperature or increase in pulse rate for different lengths of exposure in atmospheres of different effective temperatures, for different activities, expressed by their equivalent metabolic rate. It is shown that a relatively small amount of additional data to be collected in similar studies covering the necessary range in temperatures, humidities and air motion conditions will be adequate for interpreting the results over the entire range of possible human work. A program for research to supply this information is outlined.

Acknowledgment

The authors acknowledge the valuable aid rendered by Dr. T. Lyle Hazlett, Director of the Department of Industrial Hygiene of the College of Medicine of the University of Pittsburgh, in the cooperation given by his department in planning and prosecuting the physiological phases of the study. They also acknowledge the trying services of the students and others serving as sub-

jects and observers, without whom the study could not have been successfully made.

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J. K. Peacock Dies

The Society lost one of its Life Members in the sudden death of J. K. Peacock on March 7, 1939, from a heart attack. Mr. Peacock passed away at his home, 440 Fowler Ave., Pelham Manor, N. Y., where he lived with his sister, Miss Anne Peacock.

Mr. Peacock was a native of Indiana where he was born in 1868. He had a long career in the heating and ventilating industry, being associated for 15 years with J. H. McLain Co., as salesman and branch manager. Later he was with Pierce,

J. K. Peacock

Butler and Pierce Corp. working on heating problems and plans as draftsman and engineer on heating layouts. He then joined Hoffman Specialty Co., Inc., as manager of the New York office and at the time of his death he was assistant secretary of that organization in the general sales department.

In 1921 Mr. Peacock joined the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, which honored him by election to Life Membership in 1938. He was also a Life Member of New York Chapter.

Funeral services were held in Cambridge City, Ind., and the officers and Council wish to express their sincere sympathy to his family who survive, Miss Peacock of Pelham Manor and Mrs. Alice M. Ludington of Cambridge City.

CORRELATING THERMAL RESEARCH

As a part of the efforts of the ASHVE Research Laboratory to correlate research in thermal engineering carried on by the many institutions engaged in such work, and to disseminate the published results of such studies together with other reports of progress in the field, and in order to make this information available to the membership of the Society, there is published monthly on this page a limited number of brief abstracts of articles which it is believed will be of interest to all concerned. For more complete lists address the Librarian, ASHVE Research Laboratory, U. S. Bureau of Mines Experiment Station, Pittsburgh, Pa.

F. C. Houghten, *Director.*

W. L. Fleisher, *Chairman*
COMMITTEE ON RESEARCH

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Refrigeration as Applied to Air Cooling, by E. S. Green. *Journal of the Institution of Heating and Ventilating Engineers,* Vol. 6, No. 71, Jan. 1939, pp. 551-589, with discussion. Discussion of some of the variations of equipment and design conditions which can be employed in refrigerating plants which cool air used for the control of temperature and humidity of a factory, office or public building, and how they affect the initial cost and operating cost. Physical properties of common refrigerants.

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Relative Humidity with High Summer Temperatures at Boise, Idaho, by George F. Von Eschen. Published in the *Monthly Weather Review*, Weather Bureau, July, 1938. Data are compiled from records of the past seven years. Available from the Superintendent of Documents, Washington, D. C. Price 20 cents.

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Care in Installation Necessary for Good Air Cleaning, by H. C. Murphy. *Heating and Ventilating*, Vol. 35, No. 12, Dec., 1938, pp. 32-34. Suggestions for installing air filters including suitable location of filters relative to the fan, relative to heating and cooling coils, and relative to the direction of air flow. Location of air inlets away from contamination of dust or odors.

•

Orientation for Sunshine. *Architectural Forum, Products and Practice*, June, 1938, pp. 18-22. Discussion of solar radiation conditions at 30 deg, 40 deg and 50 deg North Latitude.

•

Analysis and Design of Regenerative Compressed Air Scheme for Mine Cooling, by Prof. W. J. Walker and Dr. R. L. Straszacker. *The Journal of the South African Institution of Engineers*, Vol. XXXVII, No. 2, Sept., 1938, pp. 22-40. Analysis of all important factors such as mine depth, conductivity coefficients and final temperature of discharge of the cold compressed air, on the coefficient of performance of the system. Cost of the regenerative system.

•

Automatic Heat and Its Control, by W. B. Hughes. *Coal-Heat*, Vol. 34, No. 5, Nov., 1938, pp. 5-7. Discussion of the primary fundamental requirements of a heating system that can be controlled satisfactorily: (1) generation of heat; (2) absorption of heat energy; (3) distribution of heat energy; (4) transmission of heat energy into spaces to be heated; (5) control of the system as a whole.

•

Coke as a Domestic Fuel, by P. Nicholls. *Coal-Heat*, Vol. 34, No. 6, Dec., 1938, pp. 32-33. "Coke is a good domestic fuel, clean and virtually smokeless, but it requires more care and skill from the homeowner to get the full satisfaction it is capable of giving."

•

Practical Suggestions for the Care and Operation of Stoker-Fired Heating Systems, by Carl J. Klermund. Published by *Coal-Heat Magazine*, Chicago, Ill.

•

Gravity Warm Air Heating for Tobacco Drying Barns and Cabinets. *American Artisan*, Vol. 107, No. 12, Dec., 1938, pp. 20-21. Description and diagrammatic layout of warm air heating system for drying tobacco, in a tobacco drying cabinet.

•

Modernized Hot Water Supply Makes Large Saving in Apartment Group, by D. D. Zink. *Heating and Ventilating*, Vol. 35, No. 12, Dec., 1938, pp. 23-24. Description of a domestic hot water supply system in seven apartment buildings.

Factory Building With Three-story Door Heated Without Boilers, by F. O. Jordan. *Heating and Ventilating*, Vol. 35, No. 12, Dec., 1938, pp. 15-17. Description of hot-blast heating in the factory assembly room of an aircraft factory.

•

Engineering Terminology, by Victor Brown and Delmar G. Runner. Published by Gillette Publishing Co., 330 S. Wells St., Chicago, Ill.; 310 pages.

•

Modern Furnace Technology, by H. Etherington. Published by Charles Griffin and Co., Ltd., London, 1938. Price 30s net. This book is essentially a text book for a student. It is extremely thorough, has tabulated data and many helpful sketches.

•

Velocity Profiles and Flow of Fluids Through a Contracted Pipe Line, by F. V. Engel and A. J. Davies. *The Engineer* (London), Vol. CLXVI, No. 4329, Dec. 30, 1938, pp. 720-722. Mathematical discussion of flow of fluids through a contracted pipe line.

•

Air Conditioning of Cinemas, by L. W. J. Henton. *Journal of the Institution of Heating and Ventilating Engineers* (London), Vol. 6, No. 70, Dec., 1938, pp. 495-539. Discussion of theater ventilation from a practical viewpoint.

•

The Engineering Aspects of Air Conditioning for Human Comfort, by Charles S. Leopold. *Ice and Refrigeration*, Vol. 96, No. 1, Jan., 1939, pp. 25-27. An engineering study of an air conditioning problem. Discussion of air conditioning equipment, including control of temperature, separate means of heating and dehydrating, *run-around* circuit of cooling and washing air, types of air conditioning installation.

•

How to Make a Simple Carbon Monoxide Estimator. *Heating and Ventilating*, Vol. 35, No. 12, Dec., 1938, p. 39. Description and diagram of plan for making a carbon monoxide estimator.

•

Surface Water and City Water Temperatures During the Summer. *Heating and Ventilating*, Vol. 35, No. 12, Dec., 1938, pp. 35-37. United States map of probable maximum temperatures of surface water; also table of surface water and city water temperatures in the summer in the larger cities in each state.

•

Free Air—Ventilation for Basement Parking Spaces Presents Unique Problems of Layout and Design. *Domestic Engineering*, Vol. 153, No. 1, Jan., 1939, pp. 48-49. Discussion of ventilation problems in design of basement parking spaces. Illustration of ventilation in existing basement to make it available for garage space. Duct design and amounts of clean air required for each duct opening.

•

Temperature and Pulse of White People in the Tropics. Foreign Letter in *Journal of the American Medical Association*, Vol. 111, No. 22, Nov. 26, 1938, p. 2029. Report of some data taken by Dr. Semmola in Somaliland. Temperature and pulse determinations were made of 600 normally healthy Italian soldiers in Somaliland.

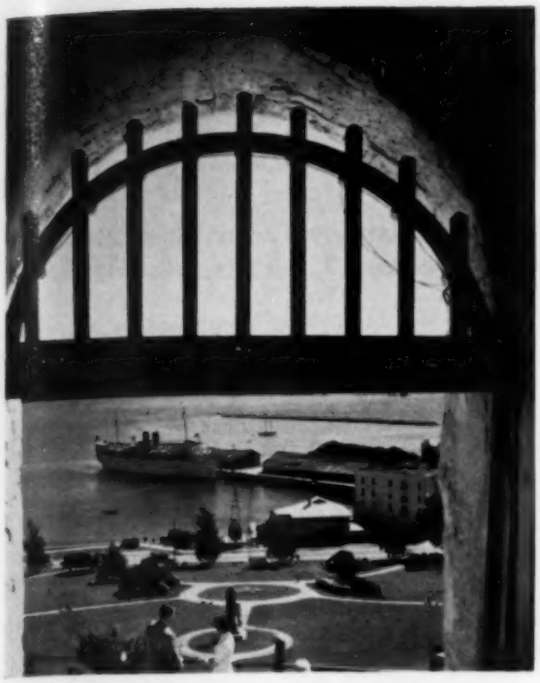
•

With No Windows the Air Conditioning Must Be Good, by J. H. Walker. *Heating and Ventilating*, Vol. 35, No. 8, Aug. 1938, pp. 13-16. Discussion of problems encountered in construction of new windowless office building. Two factors, adequate temperature control and omission of drafts, were made necessary by the windowless glass block construction.

•

Instantaneous Measurement of Temperature. *The Fuel Economist*, Vol. XIV, No. 154, July 1938, pp. 239-262. Description of a new recording optical pyrometer that is used for exposures like a camera and may be focused on any point of the surface of a hot body.

Mackinac Meeting in July



View of Harbor from Old Fort Mackinac

THE Michigan and Western Michigan Chapters of the Society will be hosts for the Society's Semi-Annual Meeting 1939, which will be held at Mackinac Island, Mich., July 4-6. Headquarters will be at the Grand Hotel and the program for this meeting, to be known as the Great Lakes Summer Meeting, is well under way. Assisting the Committee on Arrangements are representatives from each of the Chapters bordering the Great Lakes.

Mackinac Island is a unique island resort with a wealth of scenic beauty and much of historic interest to attract the visitor. Here in this delightful spot members can enjoy interesting technical sessions and participate in their favorite summer recreational activities.

The headquarters hotel, The Grand, situated on a bluff overlooking the Straits of Mackinac, has every facility for entertaining its guests and is the world's largest summer resort hotel. Special rates (American plan including room and three meals) are available to members as follows:

Double Rooms, with bath, (rear)—$8.00 daily per person.
Double Rooms, with bath, (lakeview)—$9.00 daily per person.
Double Rooms, with bath connecting, (rear)—$8.00 daily per person.
Double Rooms, with bath connecting, (lakeview)—$9.00 daily per person.
Single Rooms, with bath, (rear)—$9.00 daily per person.
Single Rooms, with bath connecting—$9.00 daily per person.
Double Rooms, without bath—$6.50 daily per person.
Single Rooms, without bath—$6.50 daily per person.
European Plan rates (room only—no meals) deduct $2.50 per day per person.

The transportation facilities to Mackinac are excellent—good roads for motorists, through Pullman car service and direct boat service.

Mackinac Island is known as the "Bermuda of the North," and does not permit automobiles on the island, but motorists may drive to the nearest convenient port, Mackinaw City, or St. Ignace, and take a ferry to the island.

The Michigan Central and Pennsylvania Railroads offer through car service to Mackinaw City and connecting boat service to the island is provided.

Boat service direct to the island is provided by the Chicago, Duluth and Georgian Bay Line and members in the more distant cities can motor or go by train to any of the following lake ports served by the line: Buffalo, Cleveland, Detroit, Milwaukee or Chicago.

Members who plan to attend the meeting as a part of their summer vacation may enjoy a cruise on the Great Lakes at nominal expense. It is also possible to include Mackinac in the special World's Fair circle tour tickets good from any city in the United States.

The boat schedule selected for the convenience of members is as follows:

S.S. North American

Lv. Chicago	July 1	4.30 p. m.
Ar. Mackinac	July 2	3.00 p. m.
Lv. Mackinac	July 2	4.30 p. m.
Ar. Owen Sound, Ont.	July 3	10.00 a. m.
(Transfer to *S.S. South American*)		
Lv. Owen Sound, Ont.	July 3	2.00 p. m.
Ar. Mackinac	July 4	7.00 a. m.

S.S. South American

Lv. Buffalo	July 1	12.00 noon
Lv. Cleveland	July 1	12.00 m.
Lv. Detroit	July 2	10.30 a. m.
Ar. Owen Sound, Ont.	July 3	10.00 a. m.
Lv. Owen Sound, Ont.	July 3	2.00 p. m.
Ar. Mackinac Island	July 4	7.00 a. m.

Typical rail and boat transportation costs from Great Lakes Chapter cities are as follows:

	Minimum Round Trip—Boat with Berth and Meals	Round Trip—Rail (21-day limit) and Lower Berth Pullman
Buffalo	$54.00	$37.90
Cleveland	43.50	32.35
Chicago (Direct)	22.50	24.00
Chicago (via Georgian Bay)	46.50	...
Detroit	37.50	21.60

Fare on the ship includes transportation, state room, and meals. (Rates are somewhat higher if deluxe accommodations are selected.)

The Committee on Arrangements hopes to provide a summer

The Old Fort—Mackinac Island

meeting of outstanding interest and enjoyment. Working under the direction of the General Chairman, W. G. Boales, are F. J. Feely of Detroit, who will be Chairman of Activities on board the S.S. South American; O. D. Marshall of Grand Rapids, who will be Chairman of Activities on Mackinac Island, and Tom Brown of the Illinois Chapter, who will be Chairman of Activities on the S.S. North American.

As detailed plans are completed, members will be advised through later issues of the JOURNAL, and by special bulletins.

Plans should be made now to attend the Great Lakes Summer Meeting at Mackinac Island, July 4-6.

Pres. McIntire Visits Chapters

A series of meetings of engineers, architects and contractors interested in air conditioning will be addressed by J. F. McIntire, Detroit, president of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

Mr. McIntire will speak on The Birth of Air Conditioning and Its Future. In his talk, Mr. McIntire notes that the great analytical mind of Benjamin Franklin first called attention to this popular subject in America, but progress in the art was slow for 150 years. Some of Franklin's contributions to comfort were the Franklin stove, chimney specifications which are substantially those used today for proper draft, proof that air motion caused cooling by evaporation, and that white clothes instead of black should be worn in a hot, sunny climate.

Modern air conditioning, according to Mr. McIntire, commenced in the Victorian era and was first referred to by that name in a book by Stuart Cramer. The art reached a high stage of development for industrial uses, but did not attract public attention until the early 1920's when the new application to comfort was announced.

At the present time the importance of air conditioning to health is engaging much attention and a substantial part of the Research activities sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS. It is Mr. McIntire's prediction that these extensive researches, conducted at the Society's Laboratory at the U. S. Bureau of Mines, Pittsburgh, and in 12 cooperating universities and colleges will result in better quality products, greater comfort, and longer life.

Mr. McIntire addressed a regular meeting of the Atlanta, Ga. Chapter of the Society, March 20, luncheon meetings at Birmingham, Ala., March 21, New Orleans, March 23, and Houston, Tex., March 25, as well as members of the faculty and student body of A. & M. College of Texas.

He attended regular meetings of the North Texas Chapter, Dallas, March 27, Oklahoma Chapter, Oklahoma City, March 29, and his itinerary includes Southern California Chapter, Los Angeles, April 11, Golden Gate Chapter, San Francisco, April 18, Pacific Northwest Chapter, Seattle, May 1, and Illinois Chapter, Chicago, May 15.

ASRE Food Preservation Conference at University of Texas

Plans for the Third Food Preservation Conference, to be held at the University of Texas, Austin, on April 13-14, under the auspices of the University with the cooperation of the *American Society of Refrigerating Engineers*, the A. & M. College of Texas and the *Southwestern Ice Manufacturers Association*, have been completed. A large attendance is anticipated as interest in the subject of food refrigeration, especially in quick freezing and farmers' lockers, has grown rapidly in the last year.

The chairman of the conference is Byron Short, professor of mechanical engineering, University of Texas, and his committee includes C. J. Eckhardt, V. L. Doughtie, Jennie Wilmot, V. T. Schuhardt and Dean W. R. Woolrich, of the University; G. E. Hulse, D. L. Fiske, M. L. Diver and T. W. Carraway, of the *ASRE*, L. P. Reiss of the *SIMA* and C. W. Crawford of A. & M. College of Texas.

Registration will begin at 9:00 a.m. on April 13 and five sessions will be held devoted to the topics of Food Technology, Locker Storage Plants, Ice Freezing and Melting, Quick Freezing, and Refrigerated Transport.

A welcome luncheon is scheduled for April 13 at 12:30 p.m. and a dinner at 6:30 p.m. on April 14. On Saturday, April 15, an inspection trip will be made to Marshall Ford Dam.

ASTM Meeting Discusses Thermal Insulation

The *American Society for Testing Materials* held a regional meeting in Columbus, Ohio, on March 8, 1939, at which J. H. Walker, Detroit, was the presiding officer at one of the sessions and F. C. Houghten, Pittsburgh, was a speaker and discussed the Effect of Solar Radiation on the Heat Transmission Through Walls.

Morning and afternoon sessions were devoted to a symposium on Lime, while a second afternoon symposium covered Thermal Insulating Materials.

Western Michigan Chapter Meets at Kalamazoo

February 20, 1939. A meeting of Western Michigan Chapter was held at Walwood Hall, Western State Teachers College in Kalamazoo, Mich. Dinner was served at 6:45 p.m. to 19 members and 23 guests.

Pres. C. R. McConner called the meeting to order and after roll call by Secy. W. G. Schlichting, minutes of the January meeting were read and approved.

Attention of the Chapter was called to the suggested ordinance prepared by the National Board of Fire Underwriters, which will be offered to municipalities for enactment. This matter was referred to the Chapter's Heating, Ventilating and Air Conditioning Code Committee, which was directed to review the subject and report at the next regular meeting.

W. G. Boales, Detroit, general chairman of the Great Lakes Summer Meeting, was present and outlined further plans and arrangements which have been made.

S. H. Downs, L. G. Miller, W. W. Bradfield and B. F. McLouth, who attended the Annual Meeting of the Society in Pittsburgh, reported on and discussed papers that were presented. A short discussion followed before the meeting adjourned at 10:00 p.m.

Southern California Chapter Tours Broadcasting Studio

February 14, 1939. Southern California Chapter met in the studios of the National Broadcasting System, Hollywood, Calif., with 76 members and guests present, according to the report of Secy. A. J. Hess.

After a short business meeting, the entire group of Chapter members with their guests enjoyed an interesting and educational visit through the studios under the guidance of competent members of the staff provided by NBC.

Air Conditioning in Industry Subject of Massachusetts Meeting

February 14, 1939. Massachusetts Chapter members and guests met at Massachusetts Institute of Technology and Vice-Pres. R. M. Nee presided.

Past Pres. W. T. Jones was asked to give an account of the recent Annual Meeting of the Society in Pittsburgh, which he did in his usual pleasing manner.

The speaker of the evening was A. E. Stacey, Jr., New York, who spoke on Air Conditioning in Industry. Mr. Stacey told of the work that was being done by this Research Technical Ad-

visory Committee of the Society in regard to measurements, tentative standards and correlation of the large amount of data already available from the health standpoint. The aims and proposed work of the committee were explained and discussed and at the close of the meeting a number of questions were answered by Mr. Stacey. A rising vote of thanks was tendered to the speaker for his interesting talk, according to the report of Secy. H. C. Moore.

Scientific Work of Byrd Expedition Interests Illinois Chapter

January 9, 1939. The meeting of Illinois Chapter was called to order at 7:45 p. m. with 110 members and guests in attendance. In the absence of Pres. J. R. Vernon, Vice-Pres. Tom Brown presided. After the reading of the minutes was dispensed with, Mr. Brown told those present of plans for the Society's Annual Meeting in Pittsburgh and special transportation arrangements for Illinois Chapter members and guests.

Vice-President Brown asked for reports from the various standing committees and then called on R. T. Miller to introduce the speaker of the evening. Mr. Miller then turned the meeting over to Dr. T. C. Poulter, second-in-command of the Byrd expedition to the South Pole, and now director of research at Armour Institute. Dr. Poulter, whose subject was The Scientific Work of the Byrd Expedition, showed in motion pictures a complete story of this expedition to Little America from the time the party sailed from New York until they bade farewell to the frozen Antarctic regions. Throughout these very interesting pictures, the speaker enlarged upon and explained the scenes as they appeared on the screen. Methods of heating, ventilation and sanitation used by the expedition were of special interest, as were many phases of the wide variety of scientific observations and experiments that were carried on.

Secy. M. W. Bishop reports that the rapt attention given to the speaker through his discourse attested both to the interest in the subject and the excellence of his presentation. At the close of the talk there were many requests for more details of certain phases of the expedition, all of which were fully answered by Dr. Poulter, and it was felt by everyone present that his talk had been one of the most interesting and instructive programs presented to Illinois Chapter.

The meeting adjourned at 10:30 p. m.

Discussion of Controls at Atlanta Meeting

February 9, 1939. Pres. C. L. Templin called the Atlanta Chapter meeting to order at the Atlanta Biltmore Hotel at 8:00 p.m. and roll call revealed that there were 12 members and 17 visitors present. Minutes of the previous meeting were read and approved.

Reports of officers and committees were called for and Treas. C. B. Cole outlined to the meeting the cost per member from past records, which averaged about $1.20 per year for meeting notices plus 75c to 80c per year for incidental expenses such as dinners for guest speakers. He made a specific plea that the members make arrangements to take care of these expenses by prompt payment of dues.

Secy. T. T. Tucker opened for discussion the report on the 45th Annual Meeting of the Society at Pittsburgh in January. Each of the resolutions recommended by the Board of Governors was read and explained and upon motion duly made and seconded was adopted.

The guest speaker of Atlanta Chapter was John Haines, who held the attention of the audience for approximately 1½ hours with a well delivered and demonstrated talk on Controls, with particular reference to pneumatic systems. Of particular interest to the members and guests present was the building up of a control system, unit by unit, with full explanation of each item involved.

Secretary Tucker was instructed to tender the thanks of the Chapter members to Mr. Haines for his timely comments, suggestions and advice.

MEMBERS—HELP WANTED!

Only 28 men were listed as candidates for membership during the month of March and 35 in April. There should have been twice as many. Taking into account the loss of members by death and by resignation, we will need nearly 500 new members this year in order to show any net gain for the Society. Since at least three months a year are lost from the standpoint of membership work, we must have over 50 members a month for the remaining nine months to make our quota.

It is the purpose of the Membership Committee to make the Chapters and their Membership Committee representatives fully conscious of the work to be done and of how worthwhile the effort will be. A survey reveals that some Chapters have no Membership Committees, other Chapters have only a single man, still others have their committees working effectively and welcome the cooperation of the national Committee.

In the interest of promoting and continuing the great work that our Society is doing, the Committee hopes that all Chapters will "get in line" and at least present the matter to those men in the respective communities who ought to be members of our Society. They need us and we need them.

The Membership Committee will assist in every possible way.

E. K. Campbell, *Chairman*
Membership Committee, ASHVE

Washington, D. C. Chapter Plans for Joint Meeting

February 8, 1939. The fifth regular monthly meeting of the Washington, D. C., Chapter was held in the Dodge Hotel with 70 members and guests in attendance.

F. E. Spurney announced a joint meeting with the Baltimore engineers to be held in April, and then called attention to the inspection trip to the Sewerage Disposal Plant, which Pres. S. P. Eagleton stated would be well worth while.

President Eagleton praised the Pittsburgh members for the excellent manner in which they conducted the recent Annual Meeting in Pittsburgh. He reported that he had urged each member of the Council to select Washington for the 1941 Annual Meeting. He stated that Washington, should it be successful in obtaining the 1941 Meeting, would be most hospitable, and would make every effort to equal the high standards set by the Pittsburgh Chapter. He then extended an invitation to all eligible non-members present to become members of the Society. President Eagleton also pointed out that 40 per cent of the dues received from the members goes to the Research Fund, which provides for very vital work in progress continuously at the Society's Research Laboratory.

The guest speaker of the evening, F. J. Kurth, New York, discussed the subject of Air Distribution.

January 11, 1939. The January meeting of the Washington, D. C. Chapter at the Dodge Hotel drew a gathering of 20 members and guests arriving in time for dinner with an additional 10 for the meeting.

L. F. Nordine raised the question of a joint meeting with the

engineering group in Baltimore. F. A. Leser stated that he had discussed this question with several of the Baltimore members and their reaction was favorable. Mr. Spurney pointed out that the Program Committee was planning to arrange such a meeting in the Spring in connection with the trip to the Friez plant.

President Eagleton urged the members to attend the February meeting and announced the program.

After the announcement about the All-Engineers Dinner, L. B. Nye, Jr., asked for information concerning the bill for licensing engineers. T. H. Urdahl reported that Senator King was introducing the bill in Congress.

Dr. Lewis Meriam of the Brookings Institute was introduced and presented a very enlightening talk on the work and aims of the Institute, which was followed by adjournment.

December 14, 1938. The attendance at the monthly meeting was approximately 35 members and guests, who enjoyed dinner, followed by the meeting in the Dodge Hotel.

The first order of business was the reading of the November minutes which were approved as read. The secretary, E. V. Fineran, reported on the question of changing the meeting date, but the decision was to continue meeting on the second Wednesday of each month.

President Eagleton introduced Mr. Thompson of the Turbine Engineering Dept., General Electric Co., who gave an interesting talk about mercury vapor boilers and turbines. At the conclusion of Mr. Thompson's remarks the meeting was brought to a close with a rising vote of thanks to the speaker.

Committee Appointments Announced for Pittsburgh Chapter

February 13, 1939. The regular meeting of Pittsburgh Chapter was called to order at 7:45 p.m. in the private dining room of Stouffer's Restaurant by Pres. R. A. Miller. An attendance of 55 members and guests was recorded. The minutes of the January meeting were approved as read by Secy. T. F. Rockwell, who also read the financial report of the treasurer. It was moved by H. Lee Moore that the treasurer's report be tabled until final settlements had been made in connection with the Society's Annual Meeting.

Reports were in order and retiring chairman E. C. Smyers announced the program for the March meeting.

J. F. S. Collins, Jr. referred to the many letters of appreciation his committee had received from visitors to the Pittsburgh meeting.

Prof. C. M. Humphreys, who represented the Pittsburgh Chapter at the Chapter Delegates meeting, reported on the matters discussed.

President Miller announced the following appointments to standing committees for 1939:

Program Committee: G. G. Waters, *General Chairman;* R. B. Stanger, *Vice-Chairman;* F. C. McIntosh, Leighton Orr, E. T. Selig, Jr., R. W. Dickson, Jr.

Sub-Committee on Entertainment: R. B. Stanger, *Chairman;* J. E. McLean, H. E. Park.

Sub-Committee on Students: C. M. Humphreys, *Chairman;* F. C. Houghten.

Membership Committee: A. F. Nass, *Chairman;* H. A. Biber, *Vice-Chairman;* B. D. Landes, C. B. Hathaway.

Publicity Committee: J. E. Frazier, *Chairman;* A. W. Marshall.

National Activities Committee: F. C. Houghten, *Chairman.*

Who's Who Committee: M. L. Carr, *Chairman;* E. C. Smyers, *Vice-Chairman.*

Current Events in Research Committee: R. H. Heilman, *Chairman;* C. M. Humphreys, G. W. Penney.

Degree-Day Committee: D. W. Loucks, *Chairman;* E. S. Scanlon, F. A. Gunther.

Attendance Committee: F. B. Mahon, *Chairman;* C. E. Parks, H. E. Lore, W. H. Reed, III.

Executive Committee: R. A. Miller, *Chairman;* F. C. McIntosh, *Vice-Chairman;* T. F. Rockwell, J. F. S. Collins, Jr., M. L. Carr, C. M. Humphreys.

President Miller suggested that the Pittsburgh Chapter consider the possibility of paying the railroad fare to the Semi-Annual Meeting for the Chapter representative on the Nominating Committee. He further requested that no action be taken on this suggestion until after the 1939 Semi-Annual Meeting.

Secretary Rockwell read a copy of a construction code for air ducts as proposed by the *National Fire Protection Association* and the National Board of Fire Underwriters.

President Miller appointed the following committee to study the code and report at a future meeting: T. F. Rockwell, *Chairman;* C. M. Humphreys, P. C. Strauch, E. C. Smyers, F. C. McIntosh, E. H. Riesmeyer, Jr., F. B. Mahon, G. O. Weddell.

G. L. Simpson, Pittsburgh Lectro Dryer Corp., was introduced by President Miller as the speaker. Mr. Simpson's subject was Adsorption Dehumidification in Air Conditioning. His formal paper was brief and dealt chiefly with the quantities of water vapor and heat involved, the interchange of heat during adsorption and the characteristics of solid adsorbers. The paper was illustrated with slides showing several applications and the methods by which continuous operation is achieved through automatic shifting of adsorber beds for reactivation.

An extended discussion from the floor followed the conclusion of Mr. Simpson's paper and a small machine was available to demonstrate method of operation.

G. L. Wiggs and Philip Drinker Address Montreal Meetings

February 20, 1939. Montreal Chapter met at the *MAAA* with an attendance of 53 members and guests. The meeting was called to order by Chairman L. H. Laffoley, who extended a welcome to the guests. Minutes of the previous meeting were read by Secy. C. W. Johnson and approved by vote of the meeting.

Mr. Laffoley introduced the speaker of the evening, G. L. Wiggs, who discussed the subject of Hot Water Heating Developments. A vote of thanks to Mr. Wiggs was extended on behalf of the Chapter by G. L. Ballantyne.

Among the guests was W. A. Russell, Waterbury, Conn., a member of the Society's Council and Membership Committee, who gave a short talk on The Advantages of Membership in the ASHVE.

January 16, 1939. The meeting of Montreal Chapter at the *MAAA* was called to order by Pres. F. J. Friedman with an attendance of 46 members and guests. Minutes of the December meeting were read by Secy. C. W. Johnson and approved as read.

Prof. Philip Drinker, Harvard School of Public Health, Boston, Mass., was the guest speaker of the evening and was introduced by Dr. Pedley of McGill University. The subject of the illustrated lecture was the Significance and Control of Atmospheric Impurities. After the lecture an open discussion took place. Professor Drinker also gave a short discourse on his development of the iron lung and some of his experiences with its use.

A vote of thanks was extended to the speaker on behalf of the Chapter by G. L. Wiggs.

L. H. Laffoley made an announcement regarding the Society's Annual Meeting at Pittsburgh before adjournment at 10:15 p. m.

Michigan Chapter Dinner Dance

February 11, 1939. The Michigan Chapter of the ASHVE held their annual Valentine dinner dance at Dearborn Inn with 140 present including members, guests and friends.

The music was furnished by Dave Diamond and his orchestra.

Secy. G. H. Tuttle reports that the party ended at 12:30 a.m. after everybody had enjoyed a very delightful evening. F. J. Feely and G. D. Winans were co-chairmen of the committee in charge of arrangements for this pleasant social event.

CANDIDATES FOR MEMBERSHIP

The Constitution of the Society, as now amended, requires the following mode of procedure in voting on applicants for membership in the Society. All applications for membership are to be sent to the Secretary and the names of applicants and their references shall be printed in the next issue of the JOURNAL of the Society or sent to the members in other approved manner as ordered by the Council. When replies are received from references, the Candidate's application shall be submitted to and acted upon by the Committee on Admission and Advancement as soon as possible.

When the Committee on Admission and Advancement has acted favorably upon a Candidate's application and assigned his grade, the Council shall vote upon the election of the proposed Candidate for membership by letter ballot. During the past month 35 applications for membership have been received and the names of these men and their sponsors are published in the following list.

Members are requested to scrutinize the list with care. The Committee on Admission and Advancement, and in turn, the Council, urge the members to assume their share of responsibility of receiving these candidates into membership by advising the Secretary promptly of any whose eligibility for membership is in any way questioned.

All correspondence in regard to such matters is strictly confidential, and is solely for the good of the Society, which it is the duty of every member to promote.

Unless objection is made by some member by April 15, 1939, these candidates will be balloted upon by the Council. Those elected to membership will be notified by the Secretary immediately after election.

CANDIDATES	REFERENCES	
	Proposers	*Seconders*
BARNES, R. W., Htg., Vtg. and Air Cond. Contractor, San Antonio, Texas.	A. J. Rummel W. H. Badgett	F. E. Giesecke W. A. Ebert
BECKER, C. S., Br. Mgr., American Blower Corp., Milwaukee, Wis.	M. W. Bishop J. F. Tobin	H. W. Schreiber F. B. Kartoh (*Non-Member*)
CASSELL, G. E. R., Maintenance Mech., Chadoz Realty Co., New York, N. Y.	G. A. Belsky E. R. Snavely	T. W. Kunzog B. A. Jacobi
DANIEL, W. E., Jr., Partner, Messrs. E. Ashby & Co., London, England.	Thomas McDonald L. B. Miller	A. B. Newton M. P. Fedders
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CANDIDATES ELECTED

In the past issues of the JOURNAL of the Society the names of the following men were listed as Candidates for Membership. The membership grade of each Candidate has been assigned by the Committee on Admission and Advancement and balloted upon by the Council. We are now instructed by the Council to post herewith, as required by Art. B-III, Sec. 8, of the By-Laws, the following list of candidates elected:

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BUZZARD, F. H., Asst. to C. S. Leopold, Cons. Engr., Philadelphia, Pa.

FLORETH, J. J., Mgr. Air Cond. Div., Westerlin & Campbell Co., Chicago, Ill.

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JOHNS, C. F., Chief Engr., The Enterprise Fdry. Co., Ltd., Sackville, N. B., Can. (*Reinstatement*)

JOHNSTON, M. T., Mech. Engr., Walgreen Co., Chicago, Ill.

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MACHEREL, FERDINAND, Consulting Engr., Algiers, French North Africa.

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